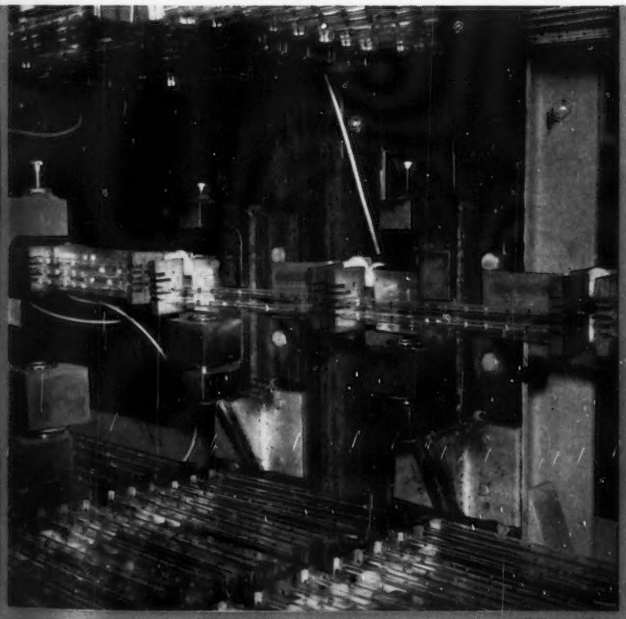
AUGUST 1961

experimental mechanics



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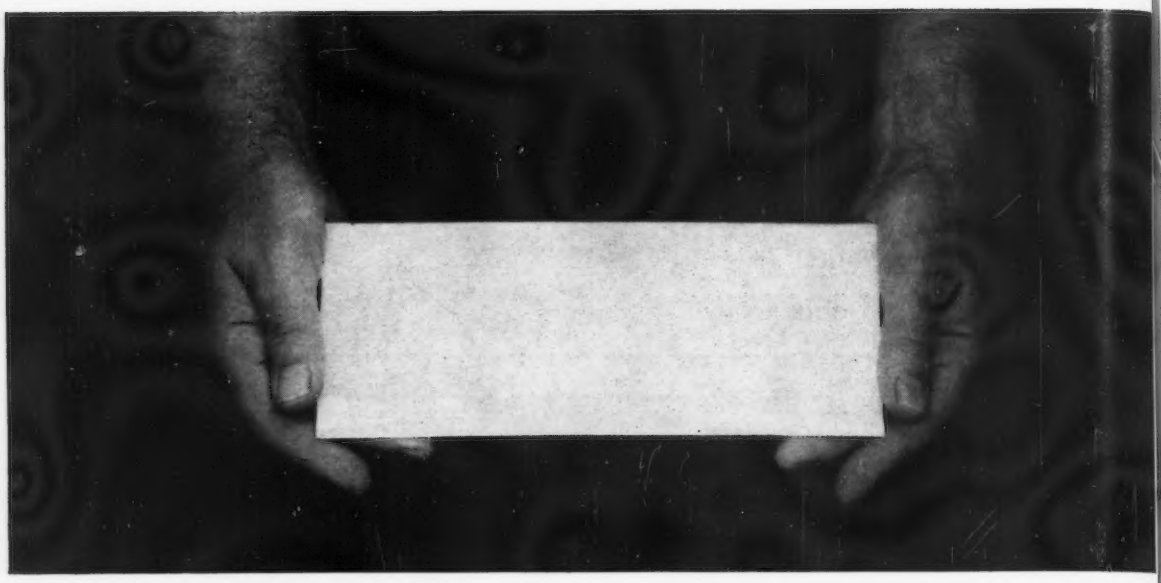
SESA JOURNAL OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS



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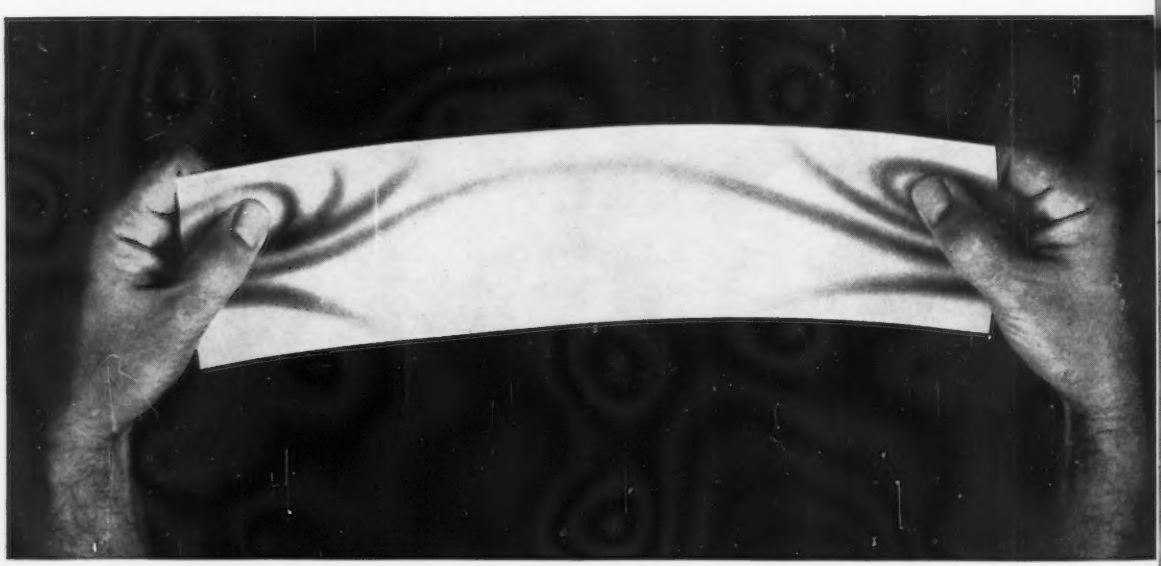

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AL OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

experimental mechanics

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SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS
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SESA PUBLICATIONS

PROCEEDINGS OF THE SOCIETY FOR EXPERIMENTAL STRESS ANALYSIS

The PROCEEDINGS are a series of bound volumes comprised of the technical papers presented at National Meetings. These contain the results and descriptions of analytical and experimental investigations, as well as techniques for the measurement of mechanical strain. Published twice annually, they include up-to-date information on various phases of instrumentation. Currently available:

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CUMULATIVE INDEX—PROCEEDINGS

Thirty-four books of the PROCEEDINGS have been published to date. This Index has been prepared to serve as an easy reference to any one of the 546 papers which have been published in the first thirty books. It consists of three parts: (1) chronological index, (2) author index, (3) topical index. It represents a "master reference file" to all PROCEEDINGS published, from Vol. I, No. 1 through Vol. XV, No. 2.

HANDBOOK OF EXPERIMENTAL STRESS ANALYSIS

This book opens with a thorough presentation of mechanical properties. It presents practical, usable information on: mechanical, electrical and optical gages, brittle coatings, photoelasticity, X-rays, service fractures, analogies, vibration measurements, theory of elasticity and precision of measurements. Stiff cloth binding—contains 1060 pages of text.

MANUAL ON EXPERIMENTAL STRESS ANALYSIS TECHNIQUES

For those who are new to the field of experimental stress analysis, the MANUAL introduces the fundamentals of experimental stress analysis in terms easily understood by the novice. It contains detailed instructions and, with the aid of illustrations and charts, elaborates on the application of three particular methods for determining stress: (1) SR-4 bonded wire strain gages, (2) stresscoat (brittle lacquer) and (3) photoelasticity.

STRAIN GAGE TECHNIQUES

Twenty-three chapters of lectures and laboratory experiments of special summer programs presented at leading universities, covering such topics as: fundamental concepts, selection of gages, basic considerations in strain-gage instrumentation, electrical and mechanical aspects of the gage system, the potentiometric circuit, the wheatstone bridge, amplitude modulation, fundamentals of measuring systems, some commercial systems and rosette analysis. Approximately 680 pages.

EXPERIMENTAL MECHANICS

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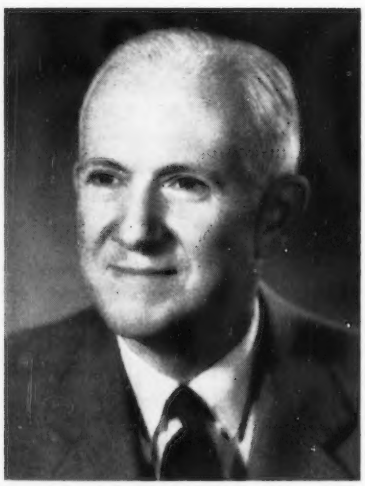
Experimentally Speaking



WHICH WAY TO GO

The world has arrived at a state which can be called complex, where nothing can stand still. In such a world what can be called progress? Its elements have always been the same, and these I shall endeavor to set forth.

As materials can be deformed in but three ways—bending, torsion and direct stress—so progress can be up, down or peculiarly sideways. The drive, ever forward and ever upward, to the so-called bigger and better, leads to the accumulation of assets, mass, speed and personnel. Heads of small businesses invariably say "We *must* get bigger."

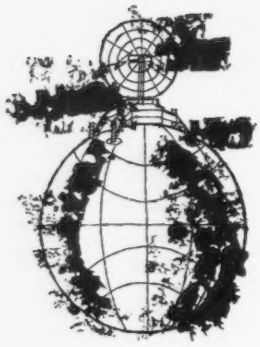
Contrariwise, to grow smaller does not necessarily mean "down and out." I have found treasures in little things and little people. Many ingenious ideas come from attic inventors. Take a little thing like the bonded strain gage. It has been said that the ideal gage is one which has no thickness or no gage length. A no-thickness gage cemented to a beam in bending lies the same distance from the neutral axis as the surface it measures and zero gage length measures strain at a point, not an average. Then, of course, there is the atom which some people deem important.

The third type of progress maintains its equilibrium without going off the chart top or bottom. It is understood by poets, philosophers and historians and by humble people. In nature and the universe, creation and destruction proceed side by side. The business cycle goes up and down. Individuals mature and decline, allowing others a day in the sun. The middle ground, maintaining equilibrium with self-compensating swings, keeps going without the pain and anxiety known to those heading for impossible terminals.

But there is something more to progress than direction, smallness or bigness. The words "bigger and better," like "ham and eggs," seem inseparable but I can prove to you that such pairs occur separately in nature. Quality is the essential ingredient of value. Without it, dimension has little meaning.

We have SESA as an example. Small but growing with care, it is faced now with a challenge to embrace new areas in the field of experimental mechanics. The question arises, "Which way to go?" The Society has done pretty well along the lines of moderation and equilibrium, with a consequent favorable quality-quantity ratio; judge this for yourself.

Francis G. Tatnall
HONORARY MEMBER, SESA



tracking station

the industry in focus

General Mills Forms Electronics Group

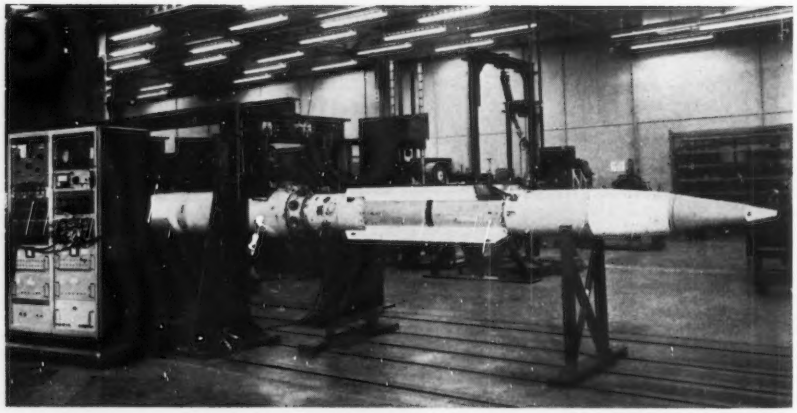
In a major realignment of operations following a period of diversification General Mills has announced the formation of the General Mills Electronics Group. The name "Mechanical Division" has been dropped and all related activities have been reorganized into four specialized departments under the Electronics Group. These are Electronic and Mechanical Defense Products, Balloon and Aerospace Systems, Automatic Handling Equipment, and Research departments. Two subsidiaries completing the Group are the Magnaflux Corp. of Chicago and the Daven Co., Livingston, N. J.

Portable Nuclear Power Plants

A knocked-down nuclear power plant was shipped via air and motor truck from The Martin Co. plant in Baltimore, Md., to 6650-ft Warren Peak near Sundance, Wyo. Sixteen 15-ton packages were hauled by C-130A aircraft to complete the airlift. Designed to operate for two years without refueling, the plant will generate 1000 kw of electricity plus 7,000,000 Btu/hr heating steam for an Air Force radar station.

The portable plant opens the possibility of power at remote sites where it is difficult to maintain supply lines for bulky fuel. Martin is under AEC contract to supply a similar plant for installation in Antarctica.

Convair Develops Missile Test Vibrators



One of several hydraulically actuated vibrators developed by engineers at Convair (Pomona) Division of General Dynamics Corp. Test requirements comprised the simulation of rough handlIng to which missiles may be subjected up to the time of their use. Sets of opposed pistons, fore and aft, provide cyclic motion in vertical, transverse and longitudinal directions. Electrical signals from the control center are programmed to produce a variety of precise vibration patterns.

High-temperature Testing Machine

This equipment makes it possible to simulate the combination of sudden, intense heat and violent pushing, pulling and twisting that places unprecedented demands upon metals used in aircraft and space vehicles. Developed at Battelle Memorial Institute, the VST (very short time) simulator can heat metal specimens to melting or to a predetermined holding temperature in a few seconds. At the same time, the specimens can be subjected to closely controlled stresses up to 400,000 psi. Technologists at the Columbus, Ohio, research center report that tensile strength, rapid heating, creep and relaxation can be studied at temperatures well above 2000° F.

Airpax Acquires Deeco

At the annual meeting of Airpax stockholders held in Cambridge, Md., April 25, approval was given for the acquisition of Deeco Instruments, Inc., a California corporation specializing in filters and toroidal components.

The Deeco Division will continue to operate under the leadership of Dean Fullmer, who became an Airpax director and a vice-president. Extension of sales, engineering and production under the Airpax name will start immediately.

Fairchild Changes Division Names

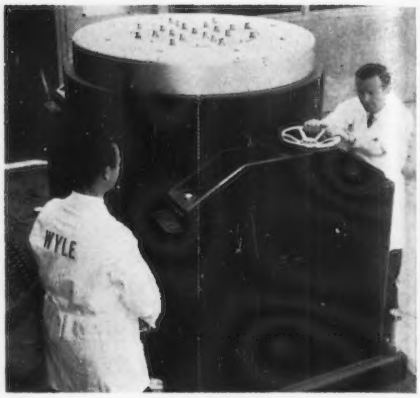
The Fairchild Stratos Corp., which recently changed its name from Fairchild Engine and Airplane Corp., has changed the name of its Astrionics Division to Electronic Systems Division. The division has its plant at Wyandanch, L. I. At the same time it announced divisional status for its St. Augustine, Fla., operation which now is designated the Aircraft Service Division. The other divisions of the corporation are the Aircraft-Missiles Division, of Hagerstown, Md., and the Stratos Division with plants on Long Island and at Manhattan Beach, Calif.

Stresscoat School Opens

The first Stresscoat Technicians School conducted by Magnaflux Corp. was a one-week lecture and laboratory course held at East Hartford, Conn. The lecturers, seated 2nd, 3rd and 4th from the left, are, respectively, Greer Ellis, stresscoat inventor; Ferdi Stern, Magnaflux field engineer; and Professor W. M. Murray of MIT who demonstrated strain gages and photoelasticity. A second course is planned for the near future.

Electro-dynamic Vibration Tester

A large shaker rated at 28,000 force pounds for random vibration and 30,000 force pounds for sinusoidal vibration has been installed at Wyle Laboratories, El Segundo, Calif. Produced by Ling Electronics, Inc., the water-cooled shaker, powered at 120 kw, is said to offer vibration testing in frequencies from 5 to 2000 cps. Higher accelerations, up to 75 g, can also be achieved. The equipment makes possible the vibration testing of larger electronic, missile and aerospace components and systems.



NLS Makes Expansion Move

Non-Linear Systems, Inc., Del Mar, Calif., has opened two new regional sales offices that are located in Denver, Colo., and Dallas, Tex., and will cover the Rocky Mountain-northwest states and south central states, respectively.

Paul R. Kemp, Jr., has been named manager of the northwest regional office, and Bob Thomas has been appointed manager of the south central region.

FUTURE MEETINGS

SESA

NOVEMBER 1–3. First International Congress on Experimental Mechanics. Hotel New Yorker, New York, N. Y.

MAY 16–18, 1962. Spring Meeting. Sheraton-Dallas Hotel, Dallas, Texas.

OCTOBER 24–26, 1962. Annual Meeting. Hotel Schroeder, Milwaukee, Wis.

- Related Conference:

OCTOBER 29–31. International Symposium on Photoelasticity. Illinois Institute of Technology, Chicago, Ill.

ASA

NOVEMBER 9–11. Meeting of the Acoustical Society of America. Cincinnati and Dayton, Ohio

IAS

OCTOBER 23–24. Institute of the Aerospace Sciences. Joint meeting with Canadian Aeronautical Institute. Quebec, Que., Can.

NRL

OCTOBER 10–12. The 30th Symposium on Shock, Vibration and Associated Environments. Detroit, Mich. Classified. Address: Code 4021 U. S. Naval Research Laboratory, Washington, 25, D. C.

SAE

SEPTEMBER 11–15. Society of Automotive Engineers. National Farm, Construction and Industrial Machinery Meeting. Milwaukee Auditorium, Milwaukee, Wis.

OCTOBER 9–13. National Aeronautic Meeting. Ambassador Hotel, Los Angeles, Calif.

personnel
in the news

STEVER ELECTED DIRECTOR

H. Guyford Stever, professor of aeronautics and astronautics at the Massachusetts Institute of Technology, was elected to the Board of Directors of Trans-Sonics, Inc.

Holder of a doctorate in physics from California Institute of Technology, and an honorary S.D. from Colgate, Dr. Stever was recently elected president of the Institute of Aerospace Sciences. On leave of absence from MIT in 1955–56, Dr. Stever served as chief scientist of the U. S. Air Force and was awarded the Exceptional Civilian Service Award for work performed during this period. Dr. Stever also received the President's Certificate of Merit in 1948 for his wartime services with the National Defense Research Council. He is presently a fellow of the American Academy of Arts and Sciences.

Dr. Stever is a member of the Defense Science Board, and chairman of the NASA Research Advisory Committee on Missiles and Space Vehicle Aerodynamics. He is also a member of the Scientific Advisory Committee to the Committee on Science and Astronautics of the U. S. House of Representatives.

WELLS IS VICE-PRESIDENT

The appointment of W. F. Wells as senior vice-president and general manager of Midwestern Instruments, Inc., was announced recently. Mr. Wells will exercise administrative control over Midwestern's engineering, manufacturing and marketing functions.

Since 1957 Mr. Wells has been manager of manufacturing engineering in the GE Computer Department at their Deep Valley Park Plant in Phoenix, Ariz.

He has a B. S. degree in mechanical engineering and in 1956 he was sent by General Electric to Harvard Business School (Graduate School Divisions) for studies in advanced marketing.

NELSON BECOMES VICE-PRESIDENT

Appointment of William E. Nelson as vice-president and director of administration of Microdot, Inc., has been announced by Robert S. Dickerman, president. As director of administration, Mr. Nelson will be responsible for the personnel, administrative services and finance and accounting departments.

Mr. Nelson was formerly general manager of Gilbert & Rothschild Co., Los Angeles realty firm, engaged in the planning and development of office buildings and shopping centers. Previously, Mr. Nelson was associated with the Union Bank for eight years. He served first as head of their legal department and subsequently as vice president and assistant to the president of Union Bank.

Mr. Nelson is a member of the American, Los Angeles and Beverly Hills Bar Associations, and is a member of the American Management Association. He was admitted to the California Bar in 1952. He is a graduate of George Pepperdine College, Loyola University Law School and the University of Wisconsin Bank Auditors and Comptrollers School.

KAHRS BECOMES MANAGER

Appointment of Gordon J. H. Kahrs as market research manager for Beckman Instruments, Inc., has been announced.

Mr. Kahrs will be responsible for expanding the company's market research programs in accordance with established growth objectives. He joins the Beckman organization after serving as a market research project director for Charles Roberts Associates, an affiliate of Ebasco Services, Inc., New York City.

TYLER HEADS TRAINING COURSES

Bruce D. Tyler will direct the five-day training courses in nondestructive testing conducted regularly by Magnaflux Corp. at their main plant in Chicago. Mr. Tyler, who joined Magnaflux in October of 1960, has had a diversity of quality control and testing experiences in both military and industrial capacities since graduating from college in 1948.

Mr. Tyler's background includes four years as an instructor-supervisor for the Air Force School of nondestructive and destructive testing, two years in Air Force Quality Control Procurement and two years with General Electric as foreman of testing and inspection. Prior to joining Magnaflux, he spent five years as instructor-writer for the Air Force Metallurgical School at Chanute AFB, during which time he studied industrial education at the University of Illinois.

COHEN NAMED MANAGER

Morton H. Cohen has been named manager of the Washington Sales District Office of Electro-Mechanical, Inc.

Mr. Cohen has a master's degree in electrical engineering from the University of Connecticut. He has previously worked for the Navy Underwater Sound Laboratories and for the New London Instrument Co. Subsequent to this, Mr. Cohen worked as a project engineer for Electro-Mechanical Research, Inc., on airborne telemetry systems, and later was a senior field engineer for Epsco, Inc., and a senior electronics engineer with the Missiles and Space Systems Division of the United Aircraft Corp. before returning to EMR as field engineer.

CARGE PROMOTED

The appointment of John Carge as manager of the Midwest District Office of Brush Instruments was announced by Norman R. Klivans, general sales manager of Brush, division of Clevite Corp., Cleveland, Ohio.

Since joining the sales organization of Brush Instruments in 1953, Mr. Carge has served in several capacities, most recently as sales engineer. He will be located at the company's main plant in Cleveland.

ZUCK MADE SALES MANAGER

Stuart H. Zuck has been appointed a district sales manager in Southern California for Non-Linear Systems, Inc., it was announced recently.

Mr. Zuck is responsible for sales of NLS's complete line of digital voltmeters, ohmmeters and other electronic instruments in the San Fernando Valley area. His office is in Pasadena.

Prior to joining NLS, he was district sales manager for Consolidated Electrodynamics Corp. Mr. Zuck received his bachelor of science degree in electrical engineering from the University of Illinos.

PARKS BECOMES SENIOR ENGINEER

Robert G. Parks has joined Datex Corp., Monrovia, Calif., as senior project engineer. He will be responsible for administration of the Heavy Industrial Systems Group.

Mr. Parks attended Wayne University before moving to California in 1952.

NINE EXECUTIVES BECOME VICE-PRESIDENTS

Nine executives of General Dynamics Corp. have been appointed to new vice-presidential positions in four new divisions of the corporation.

Mortimer Rosenbaum, chief engineer since 1957, has been named Astronautics division executive vice president. Wallace W. Withee, formerly senior assistant chief engineer, is now vice-president—research and engineering, and Elwood D. Bryant, formerly operations manager, is the new vice-president—operations at General Dynamics/Astronautics.

Also at San Diego, Calif., General Dynamics/Convair announced appointment of J. H. Famme, a 25-year veteran with Convair and predecessor companies, as vice-president—operations. Famme was formerly assistant division manager—operations.

E. R. Peterson, formerly chief of the Army Weapon Systems Department at General Dynamics/Pomona and a veteran of 20 years with Convair, was named vice-president—engineering for General Dynamics/-Convair.

General Dynamics/Pomona announced appointment of Charles D. Perrine, Jr., formerly assistant division manager—engineering, as vice-president—research and engineering; and of G. E. Sylvester, formerly assistant chief engineer—design engineering, as vice-president—operations.

General Dynamics/Fort Worth announced promotion of Robert H. Widmer, since 1959 chief engineer, to vice-president—research and engineering; and B. G. Reed, formerly assistant manager—operations, to vice-president—operations.

ROCHETTE JOINS DATEX

Walter L. Rochette has joined the Sales Department of Datex Corp., Monrovia, Calif., as a product specialist for shaft position encoders.

In his new position he will be responsible for promotion of encoder sales to both the military and commercial markets.

JOHN MOYES LESSELLS

John Moyes Lessells, founder and chairman of the board of Lessells and Associates, Inc., and former professor at Massachusetts Institute of Technology, died on May 17, 1961, in Boston after a brief illness.

Professor Lessells was born in Dunfermline, Scotland, in 1888. He attended Heriot Watt College, Edinburgh, and received his bachelor of science in engineering from Glasgow University. After service with the British War Office and as a special engineer with Rolls Royce Ltd., he came to the United States in 1920 as manager of the Mechanics Division of the Westinghouse Research Laboratories in Pittsburgh, Pa. From 1931 to 1935 he served as manager of engineering at the Westinghouse South Philadelphia Works, and in 1935 was appointed associate professor of mechanical engineering at the Massachusetts Institute of Technology, where he served actively until his retirement in 1953.

Until 1961 Professor Lessells served as president of Lessells and Associates, Inc.; at the time of his death he was chairman of the Board of Directors and a consultant to the firm. Society memberships were numerous. He was a Fellow and Honorary Member of the American Society of Mechanical Engineers, which he served for many years as technical editor of the *Journal of Applied Mechanics*. Other Societies include the American Society for Metals, The Institution of Mechanical Engineers, The American Society for Testing Materials, The Society of Automotive Engineers, American Academy of Arts and Sciences, The Iron and Steel Institute and The Society for Experimental Stress Analysis. Among his awards were the Bernard Hall Prize (IME), 1926, and the Levy Medal of the Franklin Institute in 1941 (with Dr. C. W. MacGregor).

In 1925 Professor Lessells coauthored the book *Applied Elasticity* with S. Timoshenko. He was editor of the *Stephen Timoshenko Anniversary Volume*, published in 1938, the author of *Strength and Resistance of Metals*, published in 1954. He also authored numerous papers on the fatigue of metals.

CRISWELL MADE SALES MANAGER

F. V. Criswell has been appointed sales manager for the Lerco Division of Microdot, Inc., it was announced recently.

Mr. Criswell will be responsible for sales of the Lerco line of electronic hardware, including terminals, terminal boards, handles, Mil Spec knobs and custom molded products.

Mr. Criswell was formerly sales manager of Viking Industries, Inc., connector manufacturer located in Canoga Park. Previous to his more than eight years with Viking, he was a components engineer in the standards group of Hughes Aircraft Co.

KELLER IS SALES MANAGER

George R. Keller has joined the Helipot Division of Beckman Instruments, Inc., as sales manager for Special Products. In this capacity, Mr. Keller will work closely with manufacturers of servo-systems, providing them with custom designed translatory potentiometers, linear variable-differential transformers and velocity pickups.

Mr. Keller's most recent position was president and general manager of Lido Transducers in Costa Mesa, Calif.

THOMSON MADE MANAGER

Ian F. Thomson has been appointed manager of components assembly at Non-Linear Systems, Inc.

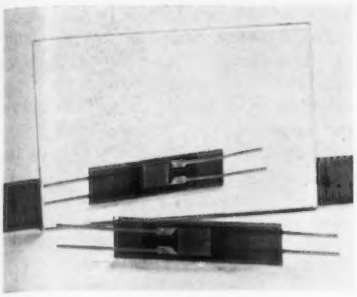
Mr. Thomson will direct development and manufacture of items used in the digital voltmeters and other electronic instruments manufactured by NLS. Prior to joining Non-Linear Systems, Mr. Thomson was senior design group engineer at Convair-Astronautics.

WELLS APPOINTED MANAGER

Appointment of D. J. Wells as sales manager of the Magnetics Division of Microdot, Inc., has been announced. Mr. Wells will be responsible for sales of transformers, magnetic amplifiers, power supplies and static inverters.

Formerly manager of West Coast sales and repair operations for Servomechanisms, Inc., he has also been sales manager for Jewett Co. and sales manager for PCA Electronics Inc.

Mr. Wells received his MBA from Stanford Graduate Business School in 1951 after completing his B.A. in economics at the University of California, Santa Barbara. He also studied electrical engineering at Caltech and served in the U. S. Navy radar program.



BENDING-SEPARATOR GAGE

A new bending-separator gage has been put on the market by the Instruments Division of The Budd Co., P.O. Box 245, Pheonixville, Pa. It is designed to separate and identify tensile strains and strains produced by bending moments on the surface of a structure.

Major feature of the new device, called "Flexagage," is that it can be mounted on one side of a structure only, thus eliminating the need for mounting strain gages back-to-back both inside and outside of such structures as pressure vessels, aircraft wings, etc.

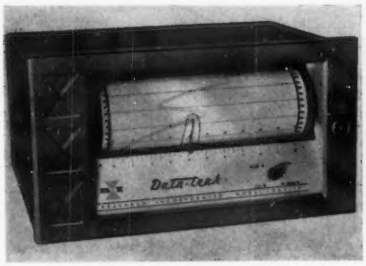
"Flexagages" may be mounted with any solvent-free, room-temperature, curing strain-gage adhesive. They are currently available in three models and can be used with materials ranging in thickness from 0.040 to 0.320 in. Standard model gages can be supplied with special correction factors for use with other material thicknesses. **25**

VHF-UHF POWER OSCILLATORS

A line of VHF-UHF power oscillators featuring compactly designed RF cavities with frequency ranges from 200 to 1050 mc and power output from 50 mw to 50 w has been announced by Microdot Inc., 220 Pasadena Ave., South Pasadena, Calif.

The Microdot units reportedly can be used for antenna evaluation, calibration of power-measuring devices, driving of amplifiers and solid-state varactors, and other applications requiring more power than milliwatt signal generators can provide.

These oscillators are continuously tunable over 2-to-1 frequency bands, and have a high-ratio dial featuring negligible backlash and a logging scale permitting a resettability of 0.002%, it is stated. **26**



CURVE FOLLOWER

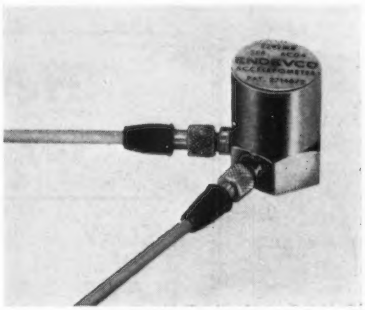
The Data-Trak function generator manufactured by Research, Inc., Box 6164, Minneapolis 24, Minn., converts manual set-point control to automatic program control. All that is required is to add a pencil-drawn program curve and to connect three wires from the Data-Trak output potentiometer to the existing circuits. As the instrument stylus follows the program, it turns a control potentiometer.

As many as four 3-gang potentiometers may be driven by the same follower mechanism to provide up to 12 separate synchronized control signals, according to the manufacturer.

The entire mechanism is enclosed in a 19-in. dust-tight case which fits a standard electronic rack. Program chart, follower probe and curve position indicators are continuously visible through the glass cabinet door. **27**

new products

For more details regarding these products, circle item number on Reader Information Card



IN-PLACE CALIBRATION ACCELEROMETER

The Model 2242M5 high-temperature accelerometer provides for in-place calibration of the telemetry system, according to its manufacturer, Endevco Corp., 161 E. California Blvd., Pasadena, Calif. A precision resistor is sealed in the base of the accelerometer so that a known current will simulate a specified g level. A charge is developed at the accelerometer proportional to its sensitivity to that a complete system checkout can be accomplished with the equipment in place.

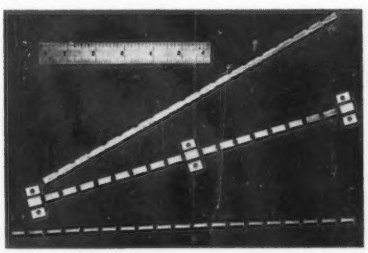
The Model 2242M5 features all-welded hermetic construction incorporating a special thermal shield to permit operation for four minutes with nearby radiant surface temperatures of 750° F. **29**

VIBRATION-DAMPING DEVICE

What is described as being an inexpensive, easy-to-apply vibration-damping device has been developed by Barry Controls Division of Barry Wright Corp., 700 Pleasant St., Watertown 72, Mass.

Known as the Strip-Damper, it is designed to minimize vibration response of such structures as air-frame components, electronic chassis, air ducts and shock and vibration test fixtures.

Weighing less than $1^1/_2$ oz., including attaching clips, the Strip-Damper is $11^3/_4$ in. long but may be cut easily to shorter lengths, with little loss of damping efficiency. **28**



X-Y STRIP-CHART RECORDER

A new, movable X-Y strip-chart recorder announced by the Tinius Olsen Testing Machine Co., Easton Rd., Willow Grove, Pa., quickly converts standard universal-testing machines into specialized units, the manufacturer relates.

Plugged into any machine equipped with an adapter unit, the recorder can be used for a wide variety of functions including stress–strain curves using electronic instrumentation; stress–time studies in which the chart is driven at a fixed speed; stress vs. specimen elongation tests using an electronic crosshead motion detector; or any other independent function provided for.

Load is plotted on the recorder in terms of percentage of selected range capacity. Because of the strip chart feature, a number of tests can be conducted in rapid sequence on the same chart or tests made on an extended time basis. A wide variety of independent functions, in addition to stress–strain and stress–time, can be recorded without affecting the routine universal-testing capabilities of the machine. **37**

SWIVEL HOOKS

New heavy-duty, ball-bearing swivel hooks for use with loads through 25 tons have been added to the line produced by W. C. Dillon & Co., Inc., 14620 Keswick St., Van Nuys, Calif.

The manufacturer states that these ball-bearing swivel hooks are especially valuable wherever cranes or hoists are required to "snake" loads into position before a vertical lift can be made. The ball bearing feature permits a full 360-deg free turn under full load and a special "knee action" feature permits an approximately 180-deg hinge movement as well. The absence of metal-to-metal bushings eliminates binding and dangerous restrictive action of heavy loads. **30**

AUTOMATIC COMPONENT EVALUATOR

Cook Technological Center division of Cook Electric Co. announces the development of the ACE-700 automatic component evaluator in answer to the many problems that arise with the rapid advance in semiconductor technology. In terms of time, manpower, cost and reliable performance, the economical and efficient test and classification of these components have been a problem to both the manufacturer and the ultimate user. Utilizing a concept of preprogrammed, plug-in test modules, the ACE-700 is said to answer this problem. The unit accepts transistors and diodes for measurement in quantities up to a maximum of 50. **42**

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Covers the general theory of bending of plates under lateral load, with applications to circular and rectangular plates, and with numerous tables simplifying the calculation of deflections and stresses in plates. Provides a treatment of combined bending and tension or compression of plates, with applications to thin plates. **By S. Timoshenko, Stanford Univ.; and S. E. Woinowsky-Krieger, Université Laval, Quebec, Canada. 2nd Ed., 580 pp., 280 illus. $15.00**

THEORY OF ELASTICITY

For engineers concerned with calculating stresses in engineering structures and machine parts—here is essential knowledge of the theory of elasticity, together with the solution of engineering problems of practical importance. The book discusses approximate and experimental methods of solving elastic problems which have proved useful in the study of complicated cases of stress distribution. **By S. Timoshenko and J. N. Goodier, Stanford Univ. 2nd Ed., 506 pp., 266 illus., $11.50**

FORMULAS FOR STRESS AND STRAIN

A compact and readily usable summary of important formulas, facts, and principles pertaining to strength of materials. This reliable reference for engineers provides helpful formulas for stress analysis and elasticity. All experimental data and empirical formulas are in line with current information. New material has been added, ensuring complete coverage of stress and strain. **By Raymond J. Roark, Univ. of Wisconsin. 3rd Ed., 381 pp., $8.25**

THE STRAIN GAGE PRIMER

What the bonded wire resistance strain gage can do in solving problems in experimental stress analysis is explained in this practical book. All phases are covered—from selecting the proper commercial gage through preparing surface of the test member, cementing, damage-proofing, and wiring the gage—to selecting the proper instrument to record the strain gage data, and interpreting readings in terms of significant stresses. **By C. C. Perry and H. R. Lissner, Wayne Univ. 280 pp., 180 illus., $6.00**

THE TESTING AND INSPECTION OF ENGINEERING MATERIALS

A practical approach to the principles and methods of materials testing, with full details on types of tests in use. Describes modern testing equipment and covers effects of variables on test results. Includes information on micro-hardness testers, non-destructive tests, low-temperature impact tests, and more. **By H. E. Davis, G. E. Troxell, Univ. of Calif.; and C. T. Wiskocil, formerly, Univ. of Calif. 2nd Ed., 431 pp., 171 illus., 42 tables, $7.50**

APPLIED ENGINEERING MECHANICS

Emphasizes the use of reasoning rather than complicated mathematics for solving engineering problems. Employing only simple mathematics, the book proceeds from easy to difficult phases of engineering mechanics, clarifying the principles advanced with numerous illustrations. Takes you step-by-step through the solutions of scores of technical problems. **By Alfred Jensen, formerly, Univ. of Washington; assisted by Harry H. Chanoweth, Univ. of Washington. 2nd Ed., 409 pp., 746 illus., $6.50**

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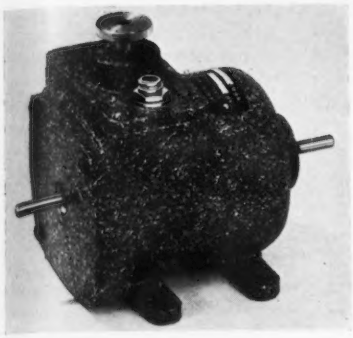


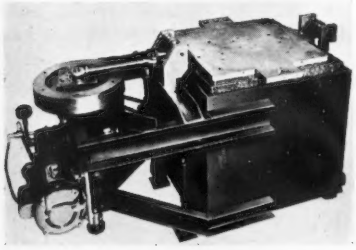
TORQUE PICKUP

A new miniature torque pickup said to be capable of accurate torque measurements in the inch-ounce area, is now available from Lebow Associates, 14857 W. Eleven Mile Rd., Oak Park 37, Mich.

The Model MTE can be used with belt or gear drive as side loading does not affect the accuracy of the torque readings, according to manufacturer.

The entire brush holder and lifter is a single assembly that can be changed by removing two screws. Brush holders are interchangeable and a spare can be installed in less than five minutes.

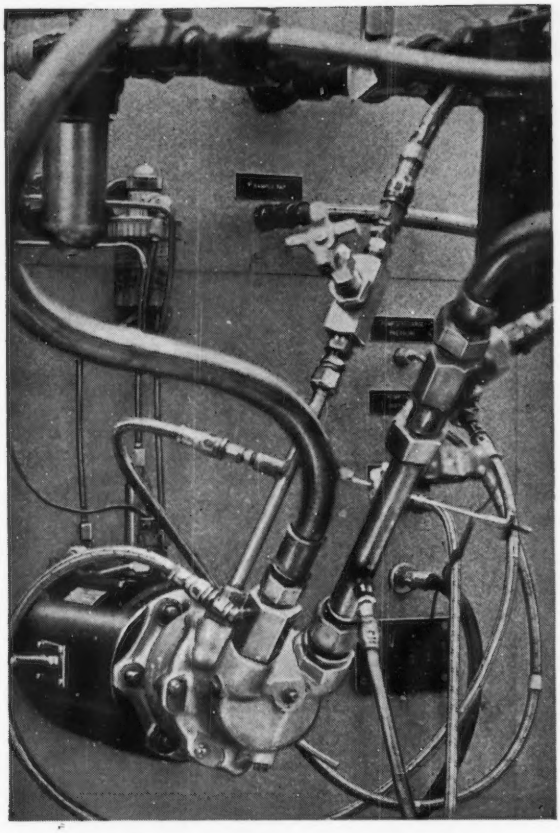
Applications include testing and design of small motors, gyros, servo motors, synchros and actuators. **47**



LONG-STROKE LOW-FREQUENCY SHAKER

Allied Research Associates, Inc., 43 Leon St., Boston 15, Mass., announces a new custom service for the design of mechanically driven vertical and/or horizontal vibration shakers. This service has been established to meet needs in industry for vibration test equipment which will operate with long strokes at low frequencies. Shakers have been designed to operate within the frequency range of 1–40 cps at amplitudes which will reach or exceed ±3 in.—an amplitude range beyond that of current vibration equipment. Other features include adaptability for vibration testing within environmental test chambers, and the accommodation of a wide range of test-package weights and sizes.

The new service is an extension of Allied Research vibration engineering activities. **49**



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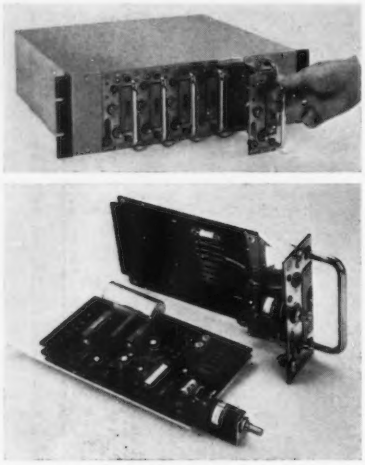
TRANSDUCER SIGNAL CONDITIONING EQUIPMENT

New units feature radically different packaging concept at low cost per channel.

Microdot has introduced two new transducer and strain gage signal conditioning units which make 0.5% or 0.1% subsystem accuracies economically feasible. Both the PS-290 Power Supply and the PB-290 Power Supply and Balance units incorporate plug-in card circuits for up to eight channels in a 19″ rack $5^1/_4$″ high.



Both units feature individual pull-out channels — bridge completion, balancing, and calibration resistors are readily accessible from the front. Lower photo shows the two separate cards that comprise a complete channel—power supply card on the left and bridge conditioning card on the right.

PS-290 Power Supply

This unit is a departure from the standard series-regulator-type power supply. New circuitry provides excellent resolution over a wide range of outputs. Warm-up time is reduced to a few minutes. Temperature drifts are negligible. Output impedance has been greatly reduced. Isolation from the line has been reduced to a level obtainable only with much more expensive equipment.

PB-290 Power Supply & Balance

This unit includes, in addition to the PS-290, the balance, calibration, and monitoring functions required in a complete transducer system. All resistors are accessible from the front of the panel and require no soldering—the power supply may be left connected and ready for use during changeover.

Write today for complete specifications and descriptions of these unusual new units.

MICRODOT INC.

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South Pasadena, California
MUrray 2-3351 SYcamore 9-9171

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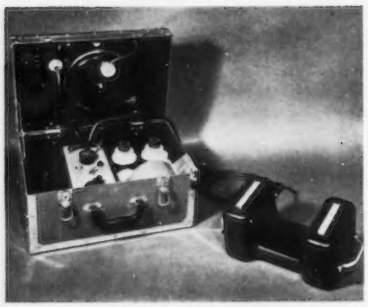


MAGNETIC-PARTICLE INSPECTION SYSTEM

"Recordaflux," a completely new magnetic-particle inspection system, has been put on the market by Instruments Division of The Budd Co., Box 245, Phoenixville, Pa.

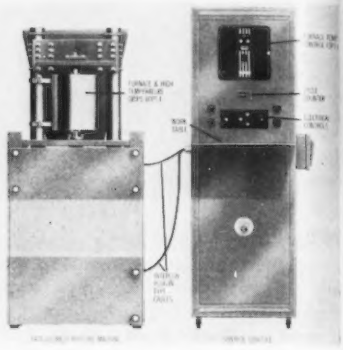
Recordaflux, formerly known as Permaweld, is used to detect defective welds, cracks, inclusions and other flaws in such materials as are currently inspected by magnetic-particle systems. The manufacturer reports that this system has been used very successfully to determine nondestructively the diameter of spot welds in stainless steel, such as cold-rolled types 301 and 201 and in precipitation hardening grades such as PH15–7MO and AM355.

The system offers to industry specially developed solutions that contain magnetic particles. When sprayed or poured on the surface of a part, the magnetic particles are free to migrate in the solution. When the solution sets up as a strippable plastic film, the magnetic particles are frozen in place. The solutions are formulated to offer good visual contrast between the particles and the part. **55**

HIGH-SPEED SENSITIVE RELAYS

Airpax Electronics Inc., Cambridge Division, Cambridge, Md., announces the availability of new high-speed switching relays utilizing a pair of magnetically operated contacts mounted in a glass tube hermetically sealed in a gaseous atmosphere. These units are rated for tens of millions of perfect operations at full load, it is claimed.

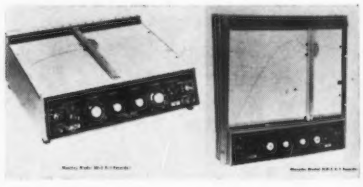
Since these relays are capable of millisecond switching time, have an operating temperature range of −55 to 150° C, and are relatively insensitive to shock and vibration, they are applicable to business machine use such as typewriters and card systems, and in computers and memory circuits, servo amplifiers, relay amplifiers and machine tools. **59**



UNIVERSAL FATIGUE MACHINE

A new Tatnall-Krouse universal testing machine for combination fatigue testing has been put on the market by the Instruments Division of The Budd Co., Box 245, Phoenixville, Pa. It is labeled Model LAZ-1.

The highlights of the LAZ-1, as reported by the manufacturer, are its ruggedness—it is stressed to withstand the most punishing testing programs; its compactness—test machine measures $29^1/_4$ x $27^1/_2$ x 67 in. high, control rack 22 x 17 x 70 in. high; and its versatility. The machine provides for static or dynamic testing on a selective basis; creep or fatigue work may be accomplished with either of the machine's two sections: The static unit (LAZ-1S), the dynamic unit (LAZ-1D), or both. **71**

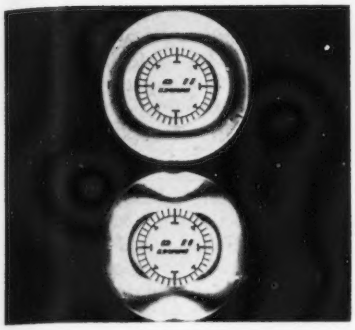


X-Y RECORDERS

Two new 11- by 17-in. X-Y recorders, each of which incorporates several unusual features yet is low in cost, are now available from the F. L. Moseley Co., 409 N. Fair Oaks Ave., Pasadena, Calif.

The new Autograf recorders are designated Models 2D-2 and 2DR-2. The latter is rack mounted; the former is a cabinet instrument which will accommodate continuous chart rolls in a variety of transport methods. In all other respects the two models are identical.

Models 2D-2 and 2DR-2 draw cartesian coordinate curves automatically from d-c electrical information, and will also plot one variable against time. **73**



PHOTOSTRESS ROSETTE GAGES

A new PhotoStress rosette gage has been put on the market by the Instruments Division of The Budd Co., Box 245, Phoenixville, Pa. It is designed to provide directly both orientation and separate magnitudes of principal strains. This important feature, coupled with the gages' speed and ease of application, promises to greatly simplify stress analysis problems for which they are used, the manufacturer reports.

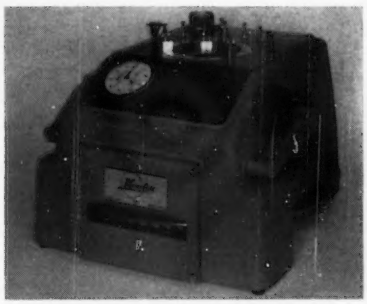
Budd's PhotoStress rosette gage has a wide range of applications. Among them are installation on such large structures as reinforced concrete or steel bridges, or on such varied big mechanical assemblies as railway and automotive frames, ocean-going ships, aircraft and large missiles. **75**

PRESSURE TRANSDUCER

A highly accurate low-range pressure transducer intended for missile applications has been introduced by the Transducer Division of Consolidated Electrodynamics Corp., a subsidiary of Bell & Howell Co, 360 Sierra Madre Villa, Pasadena, Calif.

The new Type 4-328 transducer is designed to operate in pressure ranges from 0–15 to 0–99 psi absolute. The instrument's shunt-calibration feature permits the system designer to easily provide a complete calibration of his data-handling system, according to the manufacturer.

Operable temperature range is from -100 to $+300^\circ$ F. An output of 50 mv makes the 4-328 particularly suited for use with the latest data-transmission and recording systems. Combined linearity and hysteresis of the transducer is said to be less than $\pm 0.5\%$ of full range output as measured from the best straight line through the calibration data. **77**



ANGLE POSITIONER

Kearfott Division, General Precision, Inc., 1150 McBride Ave., Little Falls, New Jersey has announced the production of the CO56402004 Precise-Angle Positioner designed for production-line operations and laboratory service. This instrument is ideally suited to any application that requires precise positioning of a rotary component, according to the manufacturer. In a typical application the precise-angle positioner is used in conjunction with a resistance bridge and null detector to measure the accuracy of synchro or resolver components. **79**

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Portable Testing Machines

Two portable testing machines—one for tension and shear tests, the other for compression and transverse loads—are described in Bulletin 60-A issued by Research Products Co., 1415 Third St. South, St. Petersburg 31, Fla.

Reported to have accuracy within ±1% the units have an operating range from 0 to 10,000 lb and weigh about 70 lb. Dimensional details, accessories available and performance data are given for the hand-operated units. **101**

Semiconductor Strain Gages

The characteristics and advantages peculiar to semiconductor strain gages are explained in 4-page Bulletin K102 offered by Kulite-Bytrex Corp., 50 Hunt St., Newton 58, Mass. Also given is a description and a tabulation of the properties of the various types of semiconductor gages manufactured by the company. **103**

Load Cells

Catalog LC-301 of the Budd Co., P.O. Box 245, Phoenixville, Pa., describes the features of load cells from the point of view of design, manufacture, specification terminology and performance characteristics. The load cell series offered by Budd Co. are identified and described in detail with clear drawings, tables and colored illustrations. **105**

SR-4 Pointer Indicator and Controller

Comprehensive information on its Type 110 SR-4 pointer indicator and indicating controller is furnished in bulletin No. 4411 published by Electronics & Instrumentation Division of Baldwin-Lima-Hamilton Corp., 42 Fourth Ave., Waltham 54, Mass.

The 6-page, 2-color bulletin provides an extensive summary on features, applications, accuracy and controls; a 2-page graphic selection chart; sample application system sketches; specifications table; dimensions and connection details.

The type 110 unit is for use in measuring and control systems which employ strain-gage transducers. Typical applications include weight and load measuring, pressure systems and torque and horsepower measuring. **107**

Remote-indicating Load-cell Systems

A 12-page engineering bulletin (R-2) offered by W. C. Dillon & Co., Inc., 14620 Keswick St., Van Nuys, Calif., illustrates many uses of remote-indicating load-cell systems throughout industry and suggests ideas for new applications not commonly known. Comparing the new Dillon differential transformer principle with conventional tube and circuitry designs, the bulletin covers 33 capacities from 0 to 100 lb through 0–1,000,000 lb, in compression, tension and push-pull models. Scientific data for in-plant engineering of complete projects is given. **109**

Data-processing Systems

Beckman Model 210 data acquisition and data-processing systems are described in an 8-page bulletin from Systems Division of Beckman Instruments, Inc., 2400 Harbor Blvd., Fullerton, Calif.

Photos, specifications, applications and detailed descriptions of the Model 210 and its subsystems are included in the bulletin. Accessories and optional peripheral equipment are discussed, as well as the means to accuracy and high repeatability in the Model 210. **111**

ASTM Publications

The American Society for Testing Materials has announced a 62-page list of publications. Published in April of this year, the list describes the symposiums, manuals, special publications, indexes, compilation of standards, charts, reference photographs and reports published by the Society through the years. More than 300 items are fully described, 40 of which are new and not previously listed.

The publications cover all phases of materials and their evaluations and are arranged conveniently by titles and subject. The list of publications may be obtained free by writing to the American Society for Testing Materials, 1916 Race St., Philadelphia 3, Pa. **112**

Weld-on Temperature Transducers

A 4-page bulletin, Special Product Note 3010, describing a complete line of platinum resistance weld-on temperature transducers is available from Trans-Sonics, Inc., P.O. Box 328, Lexington, Mass.

Having ranges with extreme limits of from −435 to +1900° F, the transducers are said to be designed to meet the severe operating environments found in missile applications. Many types are calibrated by immersion in Lox, liquid nitrogen and liquid hydrogen, as well as at the ice point. Extremely thin, the transducers present a low profile and have a fast speed of response, according to the manufacturer. **113**

Digital Data Application

"Datex Digest", a new quarterly company magazine, published by Datex Corp., 1307 S. Myrtle Ave., Monrovia, Calif., reports significant achievements in digital data recording and control. The first issue contains a survey on radiotelescope antenna position recording and control systems and other articles. **117**

Oscilloscope Camera

A pamphlet from Tektronix, Inc., P.O. Box 500, Beaverton, Ore., describes the new C-12 camera for recording oscilloscope traces.

The 4-page pamphlet gives a detailed presentation of C-12 features including one-hand portability, lift-on mounting, swing-away hinging, comfortable viewing—with or without glasses—and other details. **119**

Transducer Switching Unit

A new low-cost, modular-design transducer switching unit is described in a data sheet published by Electronics & Instrumentation Division of Baldwin-Lima-Hamilton Corp., 42 Fourth Ave., Waltham 54, Mass.

The two-color data sheet provides illustrations of the new switching unit and the Type 12 indicator, features, specifications and system drawings and prices for both a-c and d-c transducer excitation systems.

Designed for use with the B-L-H Type 12 disk indicator, the new unit is a manually selective switch which permits a reading to be taken from up to 10 transducers or transducer systems. **121**

Motion Transducers

A Special Products Bulletin covering motion transducers has just been published by Helipot Division of Beckman Instruments, Inc., Fullerton, Calif.

The bulletin points out Helipot's capabilities in the design and production of translatory potentiometers,

linear variable differential transformers and translatory velocity transducers. It includes descriptive data and illustrations of several units, as well as a listing of representative specifications. **123**

PhotoStress Technique

A 12-page manual with illustrations explains the basic principles and describes the equipment and methods used in the application of the PhotoStress technique of studying stress distributions. Step-by-step procedures given in the text are accompanied by photographs in this well-prepared bulletin PS-4000 offered by The Budd Co., P.O. Box 245, Phoenixville, Pa. **125**

Pressure Transducers

Norwood bonded strain-gage pressure transducers for precision measurement of static and dynamic pressures up to 60,000 psi, are described and illustrated in a technical bulletin issued by the American-Standard Controls Division, 5900 Trumbull Ave., Detroit 8, Mich.

The bulletin (No. 278) briefly describes the design principle upon which these devices are based, outlines their functions and advantages, and explains how air-cooled and water-cooled models operate.

Supplementary data presented includes a typical performance specifications table; a data-sheet selection chart covering 14 basic models, with maximum pressure limits for each, the variety of strain-gage bridge designs available, and threads available; a table of cable and connector assemblies; and a list of mating connectors.

Also reviewed is the Norwood Model 201 pressure monitor which, when used with a suitable two-arm transducer and an oscilloscope, provides a graphic representation of rapidly fluctuating pressures. **127**

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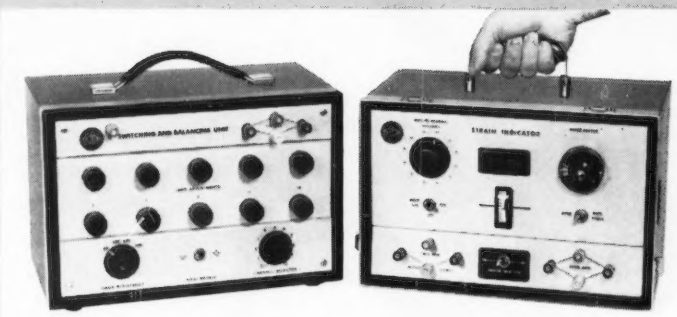

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Techniques for Testing Thermally Affected Complex Structures

Development of facilities and techniques for the experimental determination of wing deflection-influence coefficients in a thermal environment is primary objective of research program

by R. H. Gallagher, J. F. Quinn and D. Turrentine

ABSTRACT—The experimental phase of a study program relating to thermal effects on elastic deformations of built-up low-aspect-ratio wing structures is described. Techniques were developed for the experimental determination of deflection-influence coefficients for wings subjected to elevated temperatures and temperature gradients. Tests were conducted on two model wings and numerous deflection-influence coefficient sets were determined for various temperature distributions. Details of the models, the testing apparatus and instrumentation and selected test results and comparisons are given.

Introduction

In recent years, considerable attention has been given to both the analytical and experimental determination of the elastic behavior of complex aircraft structures, particularly wing structures. The geometric nature of these wings, often characterized by low aspect ratio, thick skins and complicated internal-member frameworks, precluded the use of simplified-beam-theory concepts for the determination of their elastic-deformation characteristics. Instead, it proved most satisfactory to establish a number of points on the wing planform and determine the deflections of each point resulting from the application of load to the same or another point. An influence coefficient is the deflection of such a point, known as a node point, due to a unit load application. By means of a full set of coefficients, representing the deflections of all points due to loads at all points, it is possible to describe wing deflectional behavior with an accuracy limited primarily by the number of points chosen for the representation.

The usefulness of a set of deflection-influence coefficients is due to its applicability to each of the different types of wing analysis. The structural analyst can derive, by this means, the internal load distributions for a variety of external load systems; the aeroelastician is given possession of the tools necessary for the evaluation of critical phenomena and the prediction of airloads. It is noteworthy that these analytical approaches have been made feasible only recently, coincident with requirements for more realistic analyses. The treatment of a representative number of node points is completely dependent upon the availability of high-speed automatic computational devices, a development of the past decade.

The future solution of the above-mentioned problems in a realistic manner, by means of deflection-influence coefficients, will demand a recognition of elevated-temperature effects. Aerodynamic heating of simplified wing structures is known to result in significant changes in the stiffness of these structures. Clearly, the material properties indicative of stiffness (e.g., the modulus of elasticity) are reduced in value as temperature increases. Of possibly greater significance, however, are the effects of the thermal stresses that must exist when there are nonlinear temperature gradients in a continuous structure. A review of the stiffness reducing effects of thermal-stress systems is beyond the scope of this paper, but it is sufficient to point out that compressive thermal stresses affect deflectional behavior in the same manner externally applied axial forces affect this behavior. A comprehensive treatment of the problem has been given by Kocharski and Argyris.[1]

The primary objective of the subject research program was the development of facilities and techniques for the experimental determination of wing deflection-influence coefficients in a thermal environment. Consideration of the factors involved in an actual deflection-influence coefficient determination serves to illustrate the basic difficulties. To be representative of elastic behavior throughout the entire loading range, the coefficient must be established by means of discrete load-deflection measurements encompassing this range—a time-

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consuming task. Since each value of the complete set had to be measured in an identical thermal environment, it was necessary that the environment be maintained during a testing cycle. This paper describes the facilities that have been developed to perform the function of maintaining specified temperature distributions on model wing structures as well as the means of applying loads and measuring deflections and strains in the presence of these temperatures. The critical problem of model support is also discussed. Finally, the results of tests conducted on two model wings are reviewed.

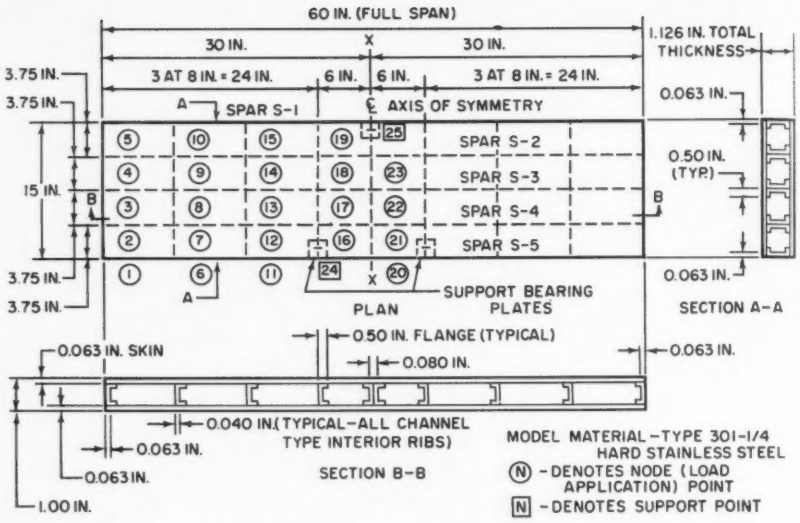


Fig. 1—First model—geometry, dimensions and node points

Models

Since the means for accomplishing the desired program objectives were in many cases a function of the details of the items being tested, the two wing models will first be described. Both models were full-span, multiweb wings fabricated of stainless steel, the first model of rectangular planform and the second model of swept planform. The reasons governing the choice of these general forms included a desire to represent realistic wing proportions and an interest in comparing the results with previous analysis and test programs.

Details of the first model are shown in Fig. 1. The 0.063-in. skins and the 0.032- and 0.063-in. formed-sheet internal members are of Type 301 ($^1/_4$ hard) stainless steel. Node points used for the determination of the wing deflectional behavior were chosen, for convenience, as the spar-rib intersection points. In testing, downward loads are simultaneously applied to corresponding points on each semispan so as to occasion deflectional symmetry about the X-X axis. This symmetry was assured at the start of testing using procedures that will be described in the section on "Instrumentation." Thus, it was necessary to measure only the deflections of the 23 node points defined by one semispan and the centerline. These points, to the left of the X-X axis, are numbered in Fig. 1.

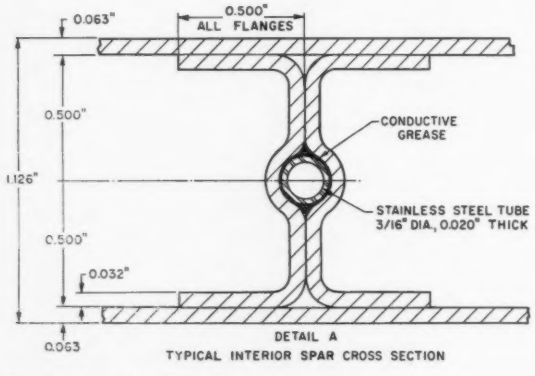


Fig. 2—Coolant-passage details

The five spars are equally spaced. The leading and trailing edge spars are single channels, while the intermediate spars are composed of two channels placed back-to-back, as shown in Fig. 2. These intermediate spar channels are formed to accommodate a coolant passage at their centerline. It was intended that, by the continuous flow of coolant through the tubes during testing, the depthwise temperature-gradients characteristic of aerodynamically heated wings could be obtained; this factor will be more completely discussed in subsequent sections. The tubes are of stainless steel, $^3/_{16}$ in. diam with 0.020-in. walls, and extend the full span of the model. To bring about improved heat-transfer characteristics, a conductive grease fills the voids between the tube and the webs within which it is contained.

The rib spacing is influenced by the support of the model at three points. These support points are symmetrically disposed on and about the spanwise centerline and delineate three rib lines as well as a "carry-through" structure. Another two ribs are placed at equal intervals between the outermost support rib and the tip rib. All ribs are single channels, except the centerline double channel, and are discontinuous, i.e., composed of pieces spanning the distance between adjacent spars.

Skins were attached to the internal structure using stainless-steel blind rivets at a pitch of approximately $^1/_2$ in. The blind riveting of both surfaces was an attempt to minimize any possible dissymmetry of structural behavior about the midplane. A coat

of black, catalytic, metal-coating paint was applied to the model exposed surfaces to provide improved heat transfer. This served to minimize the time required to heat the model to any desired temperature level as well as to maintain a uniformly consistent heat-absorptive surface.

All details of the second model were identical to those of the first—the differences between the two models being the planform and the material type. The second model is composed of Type A-286 stainless steel; its details are shown in Fig. 3.

Test Program

To separate the effects of modulus deterioration and thermal stress on stiffness, and to provide sufficient results to establish the trend of any effect over a range of parameters, several complete sets of experimental influence coefficients were determined for each model. These sets were obtained from tests at room temperature, at uniform elevated temperatures and at elevated temperatures with gradients being maintained. Since the two models were designed only for research testing requirements, exact correspondence with an assumed or actual flight plan, and its associated temperature distributions, is not possible. Rather, it was desired that realistic temperature profiles be imposed and maintained with only general requirements on the temperature magnitudes at critical locations.

The chordwise temperature profiles of aerodynamically heated wings consist of relatively high leading-edge temperatures which rapidly diminish to a value that remains essentially constant across the remainder of the chord. This type of profile was simulated by means of properly positioned radiant heat lamps. In addition, differences in geometry and in the heat transfer to the skin and internal structural members gives rise to what will be referred to as depthwise temperature gradients. During accelerated flight, these conditions result in internal member temperatures significantly lower than the temperatures of the adjacent skin, as well as local depressions in the chordwise skin-temperature profiles. The spar coolant passages served the function of depthwise gradient simulation. All depthwise temperature profiles were symmetrical about the midplane. Spanwise gradients will be encountered in a practical situation, particularly in the vicinity of the fuselage, but were not investigated in order to direct attention toward temperature gradients commonly associated with structural stiffness reduction.

Deflection-influence coefficients were first measured at room temperature (70° F) to provide the basic comparator for all subsequent testing. Deflection-influence coefficients were then measured at uniform elevated temperatures to verify that at any elevated temperature level the modification to stiffness would be of the same magnitude as the change in the value of elastic modulus. The elevated-temperature tests were approximately the same for each model. Tests were run at 350° and at 550° F, corresponding to the temperature levels of material property tests of coupons taken from the model sheet stock.

Several gradient tests were run on each model with all but the first test involving a depthwise as well as a chordwise gradient. Shown in Figs. 4 and 5 are the gradient tests for the rectangular- and swept-planform models, respectively. As can be seen from a comparison of the two figures, both the chordwise and depthwise gradients were increased for the second model. The leading-edge temperatures were established at the same level as for the first model, but the trailing-edge temperatures were significantly reduced. In addition, the depthwise gradient increases were introduced in an attempt to provide a clearer delineation of the effects of thermal stress on deflectional behavior.

The maximum values of temperature at the leading edge and points aft of the leading edge, and the maximum difference between spar average temperatures and the temperature of the adjacent skin were dictated by the upper aerodynamic performance boundaries of the study objectives. The smaller temperature levels and smaller differences represented an attempt to assess variations of stiffness reducing effects within these boundaries.

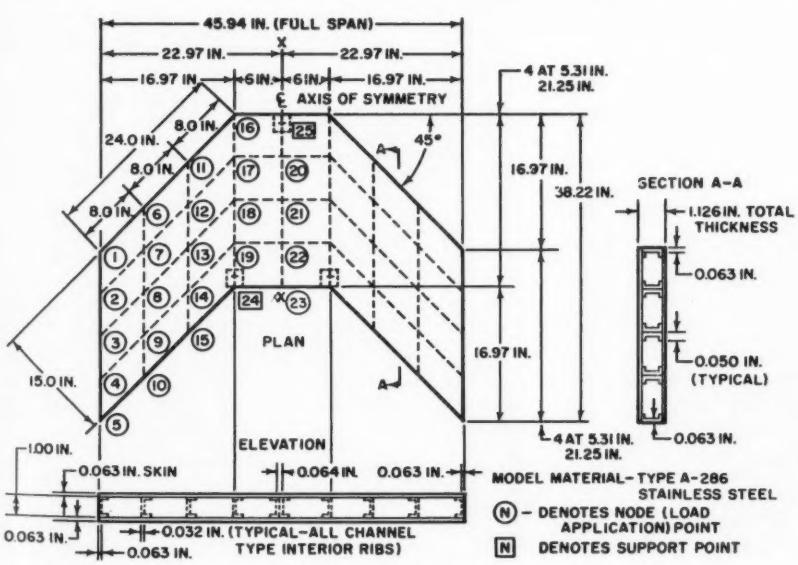


Fig. 3—Second model—geometry, dimensions and node points

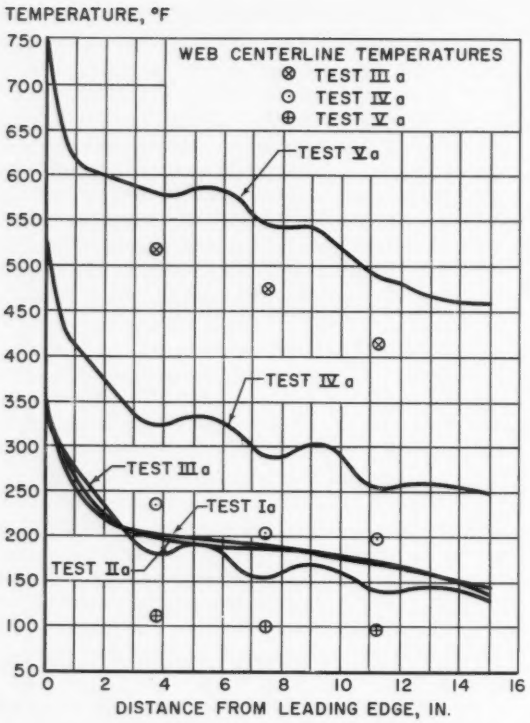


Fig. 4—Skin chordwise temperature profiles—first model

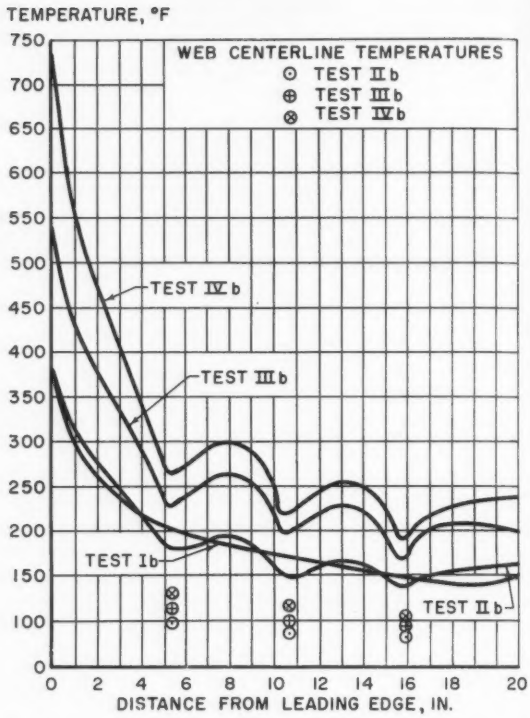
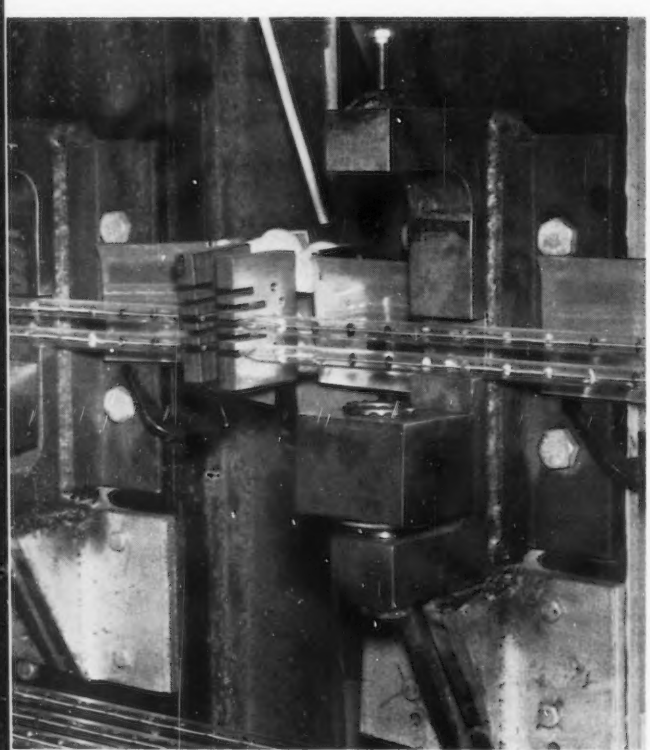


Fig. 5—Skin chordwise temperature profiles—second model



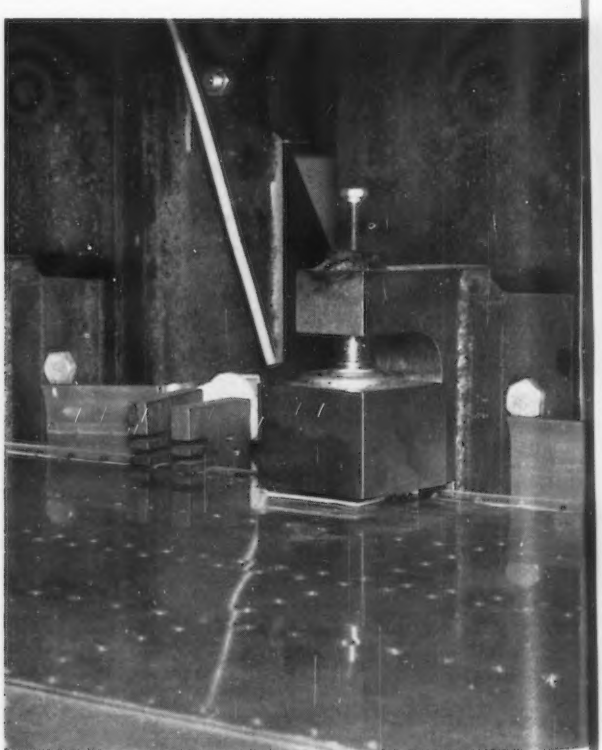


Fig. 6—Photographs of heated model-support point

Method of Model Support

One of the most critical aspects of the elevated-temperature testing of wing deformation models is the manner in which the model is supported in the loading fixture. Experience has shown that cantilever-wing deflection-influence coefficients are most satisfactorily obtained by testing full-span models with the simultaneous application of loads to symmetrically located points on each semispan.

Unheated full-span models are generally supported by clamping the model along the full length of the spanwise centerline but this procedure creates obvious difficulties when the model is to be tested at elevated temperatures. The mass of the clamp acts as a heat sink to disrupt the desired temperature distribution and this, together with the restraints exerted by the clamp, produces a complex thermal-stress system in the support area. Such a thermal-stress system will importantly affect the deflectional behavior of low-aspect-ratio wings (cf. Reference 2).

To circumvent the above-mentioned difficulties, "point" supports have been used. It has been shown that deflection-influence coefficients obtained from a model supported at three noncollinear points can be rendered applicable to a broad class of support conditions. The simulated "point" support designed for this program is shown in Fig. 6; details are shown in Fig. 7. All movable parts are contained by a C-clamp welded to a relatively massive support frame that is independent of all fixtures otherwise associated with the model (e.g., the load-application device). The top and bottom faces of the model are placed in contact with a 1-in. diam ball-bearing ring, allowing for the in-plane freedom of the model, Bearing blocks separate the ball bearing ring from spherical seats which prevent any moment restraint. Since it is also necessary to allow for vertical thermal expansion, an adjustable spring assembly is incorporated in the upper segment of the support.

The use of point supports is not without its drawbacks. For one, a true point support would result in infinitely large local stresses. The support must therefore strike a medium between reasonable local stress and a minimized area of support. Another problem is occasioned by the deflection of the supports during loading of the model. For a given loading condition, these deflections constitute the movement of the model as a rigid body. It is apparent that the differential settlement of the relatively closely spaced support points on either side of the X-X axis, representing a rotation about this axis, will produce a particularly significant error in the wing-tip node-point deflection measurements. This error is not readily taken into account since deflection measurements were normally recorded for only a single semispan.

Support-point deflections were taken into account by the following procedure: dial gages, readable to 0.0001 in., were mounted as close to each support as possible on either side in the spanwise direction. Dial gages were also mounted at the corner node points of the semispan for which deflection measurements were not normally taken. The special devices used to measure deflections in an elevated-temperature environment (to be described later in the section on "Instrumentation—Deflection Measurements") were located at each of the opposite semispan numbered node points. Readings taken from all of these instruments during the unheated tests allowed the calculation of both support and node-point deflection-influence coefficients.

The concept of support-influence coefficients, based on averaged values of the two gages to either side of the support, is considered valid because the load-deflection behavior represented by the dial-gage readings proved to be remarkably linear. More important, it was found that when rigid-body motion corrections calculated from these support-influence coefficients were applied to the measured influence coefficients of symmetrical corner node points on the two semispans, symmetrical results were obtained. This indicated that a dissymmetry of deflectional behavior about the X-X axis was primarily due to support deflection, rather than any geometric dissymmetry of the model.

Neither dial gages nor the deflection-measurement devices could be mounted close to the support during elevated-temperature testing. Rigid-body motion corrections, based on room-temperature support

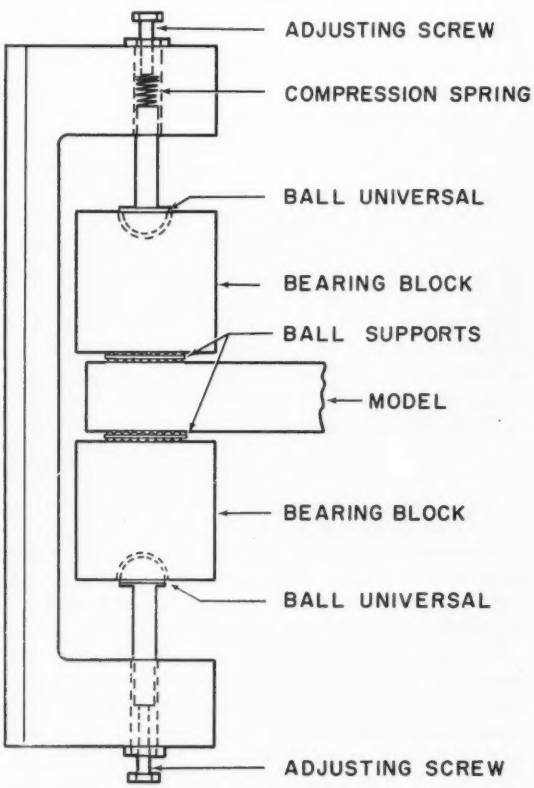


Fig. 7—Detail of point support

deflection-influence coefficients, were therefore used for the elevated-temperature test results. This is believed to have been a satisfactory procedure since the support structure was at an average temperature considerably less than that of the model and thus experienced only a slight elastic-modulus change.

Heat-transfer Facilities

Conditions of uniform temperature and chordwise gradients were established and maintained throughout each deflection-influence coefficient test using General Electric T-3 radiant heat lamps for the thermal input. These lamps were grouped in fixtures and placed, along with a number of reflector plates, above, below and around the edges of the model. The spacing between fixtures allowed the passage of loading cables and elements of the deflection-measurement devices. By dividing the complete thermal-input array in a number of sections, each consisting of a number of lamp fixtures, it was possible to control the temperature levels of several areas of the model. The output of all lamps within a section was varied using auto transformers to control the applied voltage to a section. Additional control was obtained by adjusting the distance between a fixture and the model.

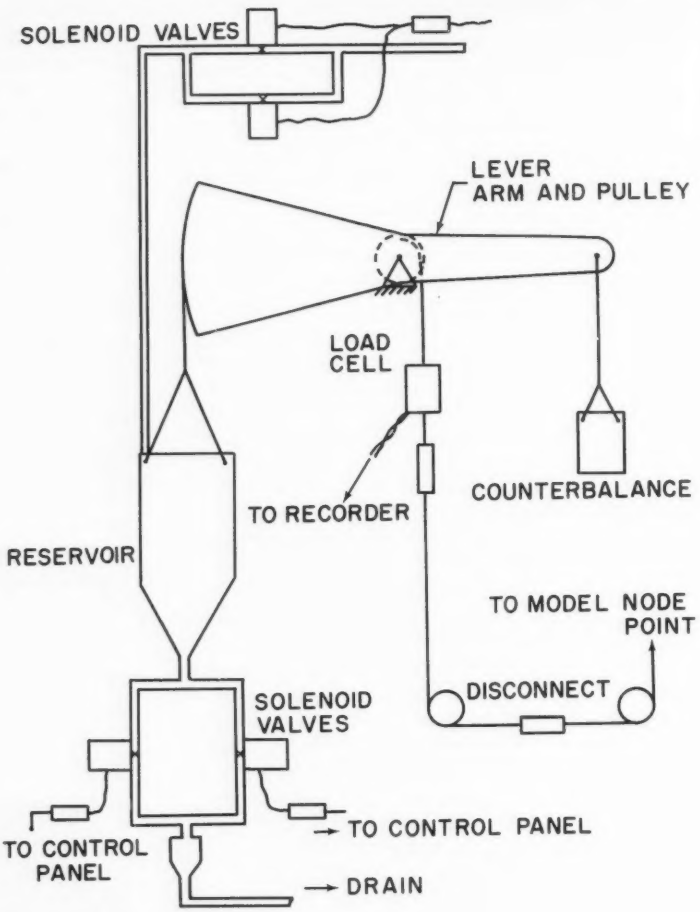


Fig. 8—Load-application system

Depthwise gradients were imposed and maintained primarily by means of coolant flow in the spar webs. The coolant-passage details have been described previously. Coolants flowing within these passages were circulated in a closed system, the choice of a coolant and many elements of the pumping system being dictated by the average temperature of the model. Water is the most convenient coolant fluid, but when used above its boiling temperature introduces many problems. Fortunately, the majority of imposed thermal environments allowed the use of water without occasioning these problems. Either ethyl glycol or air was used in the few highest-temperature gradient tests for which water could not be used. The above-noted radiant-heat input variation capabilities also contributed to depthwise gradient control.

The time required to heat up and stabilize the temperature distribution of a model was approximately three hours, with significant differences being caused by the laboratory environment, the temperature levels to be attained, etc. To stabilize the temperature distribution in the model itself, the time was considerably shorter than three hours. There was the necessity, however, to stabilize the temperatures of all items in contact with the model which would tend to produce erroneous deflection readings due to continued expansion. The support frame itself was the most likely cause of this type of error.

Method of Load Application

The apparatus used to apply load to the node points of the models, in both heated and unheated states, incorporated both hydraulic and mechanical features. The arrangement and details of the complete system are shown in Fig. 8. Since loads were simultaneously applied to symmetrically located points on each semispan, two systems of this type were required.

The static load was supplied by a volume of water confined in a cylindrical reservoir. Liquid volumes in this container were varied by means of a switching arrangement wherein two solenoid valves above the reservoir provided the alternative of either coarse or fine adjustment of the fluid input. A similar pair of valves below the reservoir performed the same function with respect to outflow.

The load, represented by the weight of water in the container, was transmitted to the model by means of a lever-arm-and-pulley arrangement, and then through a flexible-steel-cable circuit which was passed around a pair of pulleys. The lever-arm-pulley arrangement was designed so that a constant mechanical advantage existed throughout the loading range. The counterweight attached to the lever arm balanced the system when the reservoir was empty.

The flexible steel cable, referred to above, extended to an alignment pulley close to the laboratory floor, terminating at a disconnecting device located outside the heated area. Individual cables extended from each of the model node points as far as the disconnecting device, allowing the connection of any desired point to the load system. A calibrated load cell, at an intermediate location along the cable, defined the magnitude of the applied load. Measurements taken by this load cell were indicated by means of a self-balancing potentiometer.

A complete deflection-influence coefficient set in a specified thermal environment, commencing with an unheated model, was generally accomplished in a 5-hr period of time. After the model had been brought to the desired temperature profile, node-point loading was begun, proceeding with the points along the outermost rib and continuing inboard to the model centerline. Load and deflection measurements were recorded during the removal of load. The number of measurements taken between the maximum load level and no load varied with the point of load application and the model.

Instrumentation

Deflection Measurements

The need to accurately measure changes in deflection due to thermal effects required the measurement of wing deflections at temperatures in excess of 750° F to an accuracy of 0.001 in. The arrangement shown in Fig. 9 was devised to permit these measurements.

Deflection-measuring devices that can function properly in an elevated-temperature environment were not available as of the start of this program. The deflection of a node point was therefore directly transmitted from the heated area by a 10-in. long, $^1/_8$-in. diam quartz rod standing vertically on the model upper surface at the node point. The upper end of the quartz rod was connected to the metal probe of a variable-permeance extensometer,* the pivotal element of the deflection-recording system.

This extensometer consists of the metal probe, surrounded by two axially arranged coils embedded in a ceramic cement and contained within a tubular stainless-steel shell. The coils were connected by wiring to a bridge-type gaging circuit. To minimize the effects of temperature, the extensometer was centrally mounted, using transite spacers, inside a 2-in. diam aluminum container. Low-density fibrous-quartz insulation was placed between the aluminum container and the extensometer casing.

Movement of the metal probe resulted in a change in the output voltage of the bridge circuit to which the coils were wired. Voltage was provided by two CEC 2-105A oscillator power suppliers and the signal amplified using Type 1-113B carrier amplifiers. It was necessary to determine the sensitivity of the system to permit the conversion of bridge unbalance into deflection measurements. The effect of temperature on sensitivity was also determined.

After amplification, the bridge output voltage resulting from the movement of the metal probe was interpreted by a digital recorder and transmitted to a punch tape. The tape data, in turn, was transferred automatically to punched cards for subsequent use in an IBM 704 data reduction program. The digital-computer program effected the fit of a straight line to the measured load-deflection information, using a least-square criterion. This procedure resulted in the definition of the most representative deflection-influence coefficient.

Strain Gages

Since one objective of this program was the determination of thermal stress effects on wing deflectional behavior, it was necessary to define the state

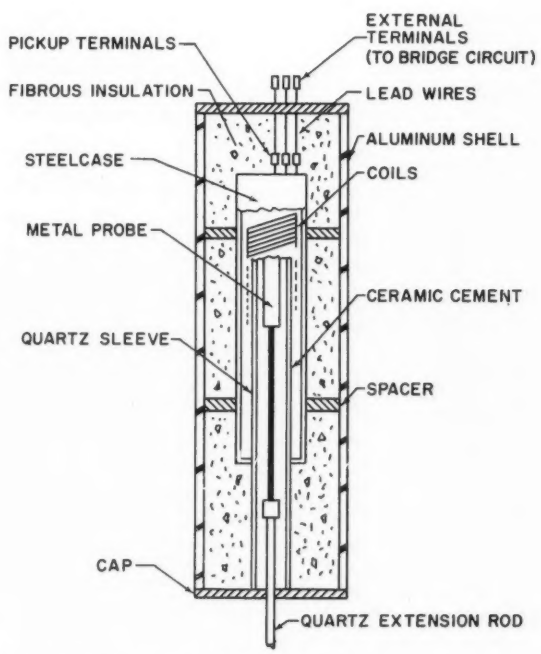


Fig. 9—Cross section of deflection pickup

* *Manufactured by the Crescent Engineering and Research Co., Pasadena, Calif.*

of stress resulting from a stabilized temperature distribution. In the case of the rectangular planform model, it was likely that computations based on thermoelastic beam theory[4] could be used to determine thermal stress. The same approach, taken with respect to the swept-wing model, was expected to be less valid.

The verification of this type of analysis was based on a comparison between the computed thermal stresses and those derived from strain-gage measurements. The majority of strain gages were concentrated on a single cross section halfway between the outermost support point and the tip on the semispan for which deflection measurements were taken. The distribution of gages on this section is as shown in Fig. 10. Other gages were located at points near the support and at the wing tips for the purpose of determining local stress effects.

The gages were generally Baldwin Type EDBF-7S, except where size limitations existed because of model geometry, in which case it was necessary to use Type AB7-5. Neither gage was expected to be useful at temperatures in excess of 400° F and, once this temperature had been exceeded, it was assumed that subsequent usefulness at any temperature level had been destroyed. All gages were mounted on the inside of the model.

The zero shift of each gage was measured by heating the entire instrumented model uniformly, in steps, to the maximum permissible level established by gage limitations. The temperature in each instance was stabilized and the indicated strain recorded. The resulting data for each of the two types of gages were determined and applied to the measured values of total strain to arrive at that portion related to thermal stress.

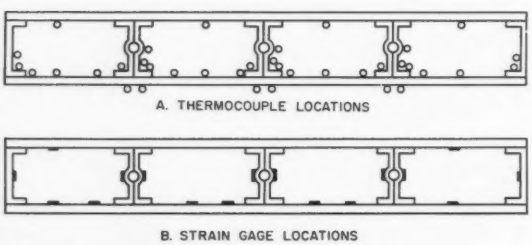


Fig. 10—Instrumentation location at reference section

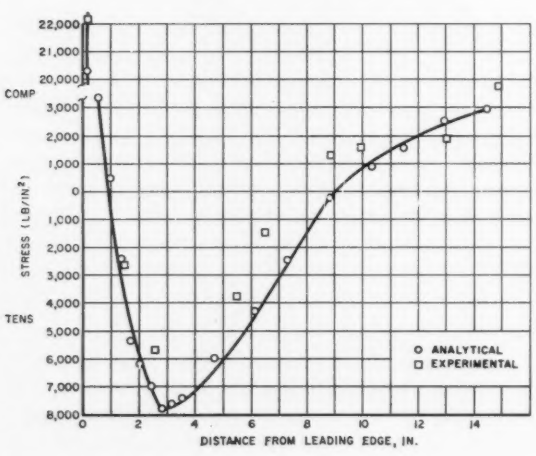


Fig. 11—Analytical versus experimental stresses

Temperature-recording Equipment

Thermocouples were attached to both the model and portions of the testing apparatus. Those affixed to the model were necessarily placed at all strain-gage locations, at the cross section used for thermal-stress computations and at various locations for temperature control purposes. Thermocouples were installed within the deflection-device enclosures to obtain information needed for computing changes in sensitivity of the deflection-measuring system, and in the coolant lines as an aid in flow regulation.

Thermocouple junctions were formed from No. 30 gage iron-constantan wire surrounded by a fibrous-glass insulation. Those few that were located on the model outer surface were coated with a reflective paint to insure values of emissivity and absorptivity lower than those of the surrounding material. The generated emf was transmitted through lead wiring to a pair of automatic switching boxes coupled to a 12-channel recording potentiometer. This arrangement permitted a continuous visual survey of model temperatures.

Results and Conclusions

The complete results of tests performed on the two model wings are presented in Reference 5. These data are quite extensive, since over 50 strain gages were mounted on each model and each deflection-influence coefficient set represents 529 distinct numbers. Nevertheless, it is possible to present some results of interest in an abbreviated form.

As indicated previously, one cross-section of each model was heavily instrumented with both thermocouples and strain gages to permit a comparison between analytically derived and experimentally determined thermal stresses. Figure 11 shows a plot of the skin-stress variation across the chord, as determined analytically for the first gradient test performed on the first model. Superimposed on this curve are the corresponding experimentally determined stresses. The agreement is seen to be generally satisfactory.

Because of the erratic nature of the individual deflection-influence coefficient test results, and to provide a concise basis of comparison of the various test results, the measured deflectional characteristics have been expressed in terms of a flexibility parameter. The parameter employed is the sum of the deflections at all node points as caused by the simultaneous loading of node points 1 through 10. (A detailed discussion of the choice of the parameter

TABLE 1—COMPARISON OF ELEVATED-TEMPERATURE DEFLECTION PARAMETERS WITH ROOM-TEMPERATURE DEFLECTION PARAMETER—FIRST MODEL

Room-temperature deflection parameter* = 3.66795 in.
Room-temperature modulus of elasticity (E) = 29.1×10^6 psi

Test	T avg., ° F.	E avg., psi, $\times 10^6$	Deflection parameter, in.	Increase of test parameter, %	Decrease in E, avg., %
Uniform temperature Ia	343	27.7	3.82854	4.38	4.81
Uniform temperature IIa	567	26.5	4.01412	9.44	8.93
Gradient Ia	198	28.4	3.81044	3.88	2.41
Gradient IIIa	179	28.5	3.86859	5.47	2.06
Gradient IVa	310	27.8	4.00635	9.23	4.47
Gradient Va	547	26.6	4.06326	10.78	8.59

* Deflection parameters for all tests equal the sum of deflections at all points for loads applied at points 1 through 10 inclusive.

TABLE 2—COMPARISON OF ELEVATED-TEMPERATURE DEFLECTION PARAMETERS WITH ROOM-TEMPERATURE DEFLECTION PARAMETER—SECOND MODEL

Room-temperature deflection parameter* = 2.11103 in.
Room-temperature modulus of elasticity (E) = 29.9×10^6 psi

Test	T avg., ° F.	E avg., psi, $\times 10^6$	Deflection parameter, in.	Increase of test parameter, %	Decrease in E, avg., %
Uniform temperature Ib	362	27.1	2.28896	8.43	9.36
Uniform temperature IIb	543	25.3	2.41753	14.52	15.38
Gradient Ib	192	28.7	2.29971	8.94	4.01
Gradient IIb	189	28.8	2.31212	9.53	3.68
Gradient IIIb	252	28.2	2.40735	14.04	5.69
Gradient IVb	296	27.7	2.43995	15.58	7.36

* Deflection parameters for all tests equal the sum of deflections at all points for loads applied at points 1 through 10 inclusive.

is given in Reference 5.) For illustrative purposes, the parameters were each compared with the room-temperature test parameter. Table 1 shows these percentage differences for the results of the tests performed on the rectangular-planform wing. Table 2 provides a similar representation for the swept-planform wing. Gradient test IIa was left out of Table 1 since it represented the same gradient condition as test Ia with the leading and trailing edges reversed.

Although the individual influence coefficients were not of a sufficient degree of consistency to lend absolute confirmation to the results of comparisons based on the flexibility parameters, certain conclusions may nevertheless be drawn. These conclusions are:

1. At any uniform elevated-temperature level, the modification to stiffness was of the same magnitude as the change in value of elastic modulus.

2. The independent effect of thermal stress on the deflectional characteristics of the two model wings did not prove to be very large on the basis of the results shown in Tables 1 and 2. The magnitude of stiffness change due to the presence of chordwise gradients is less than 5% for all tests on the first model, and less than 9% for all tests on the second model. The results demonstrate that built-up wing structures are less susceptible to the stiffness reducing effects of thermal stresses than are wing structures of high solidity, provided these stresses are within the material elastic limit and cause no skin-panel buckling.

Comparisons on a coefficient-to-coefficient basis proved to be erratic, negating the possibility of useful torsional-loading comparisons. The simplified concept of torsion, however, can be only approximately defined for structures whose elastic deflectional behavior is represented by a set of deflection-influence coefficients. More realistic assessments of thermal-stress effects would be obtained by comparing the results of analyses in which influence coefficients are normally used, e.g., natural frequency determinations. Analyses of this type have not yet been performed using the model test data.

3. The depthwise gradients that were superimposed on a chordwise temperature gradient did not significantly modify the deflection characteristics displayed by the model with the same chordwise gradient applied independently. Gradient tests I and III in Table 1, and gradient tests I and II in Table 2, are illustrative of this point.

Finally, it can be concluded that techniques are available for the experimental determination of the elastic characteristics of heated wing structures. These techniques are applicable when the characteristics are defined by means of deflection-influence coefficients and the thermal environment is a specified temperature distribution. Their feasibility was demonstrated by the series of tests performed on the two model wings.

Acknowledgment

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Photoelastic Study of the Stresses Near a Cleavage Crack

Results of analysis indicate that the higher stresses are localized in a region of the order of the half width of the test specimen in size

by R. Guernsey and J. Gilman

ABSTRACT—Photoelastic analysis has been used to determine the stress distribution near the tip of a cleavage crack. The crack bisects a slender rectangular bar along its length, and forces are applied at one end of it tending to open it out. The region of high stresses is localized in a region of the order of the half-width of the bar in size. Very near the crack tip, the stresses decrease in proportion to the inverse square root of the distance from the crack tip. The maximum gradient of the principal tensile stress lies at an angle of 70 to 80 deg from the plane of the crack. Contour maps of the principal stresses, the 45-deg shear stress and the maximum tensile stress are presented.

Introduction

By the term "cleavage crack," is meant a crack that is used to divide a body into two more or less equal parts. The crack is usually started or made to move by means of a wedge that is inserted in the mouth of the crack and struck with a hammer. Since crystals and other anisotropic materials, such as wood, are often divided in this way, the process is of some interest from the practical point of view. From the scientific viewpoint, cleavage is of interest because this type of specimen can be used conveniently to study the factors that influence crack propagation in solids. Also, cleavage can be used to make measurements of the surface energies of solids. As one of the first steps in a study of the cleavage process, it was decided to determine the stress distribution in the vicinity of the crack, and since it seemed a very difficult task to find the stress distribution analytically, this photoelastic study was made.

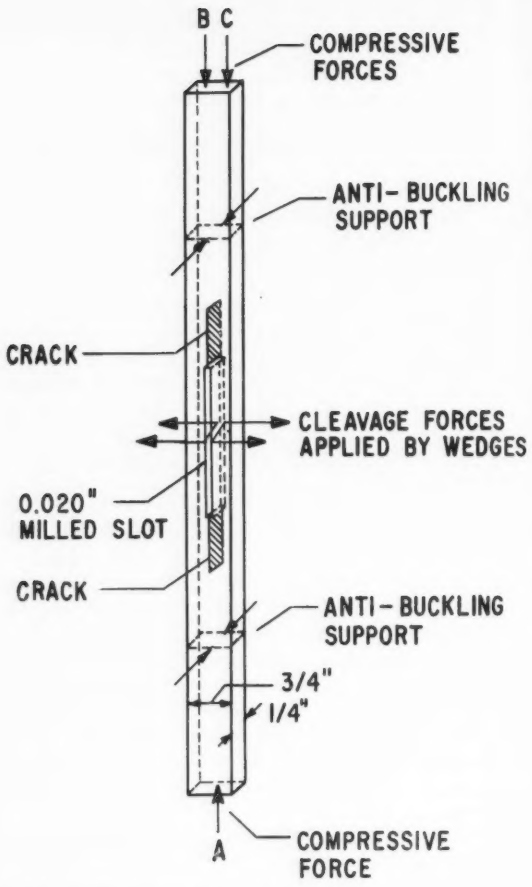

Fig. 1—Plastic model of cleavage cracks

The most difficult obstacle of the study was the preparation of the model. In cleavage-type specimens, the maximum tensile stress does not lie in a plane perpendicular to the crack, but in a plane whose normal is at an angle of roughly 80 deg to the direction of propagation of the crack. (This will become clear later.) Therefore, if the material is elastically isotropic, a cleavage crack is not expected to run straight through it. In anisotropic materials like crystals, cracks run straight because they lie parallel to the planes of weakness; but, in isotropic materials like plastics and glasses, ordinary cleavage cracks will not run straight. Since it was necessary to use an optically sensitive plastic material (Paraplex P-43) for this study, the production of a model with a straight crack in it posed a considerable problem.

The Model

After several abortive attempts, good models were made with the aid of a technique for stabilizing

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Fig. 2—Isochromatic stress fringes in cleavage-type specimen. Stresses were produced by wedging open the crack

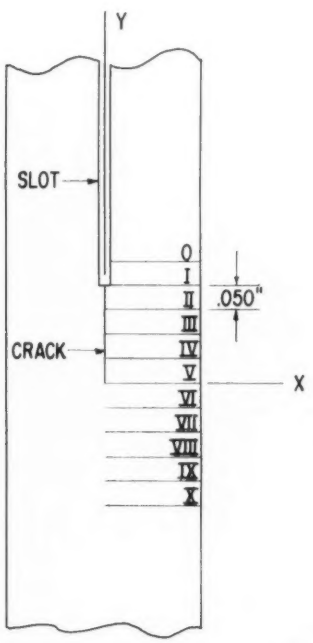

Fig. 3—Series of lines, spaced 50 mils apart, on test specimen

cracks that was conceived by Benbow and Roesler.[1] The mechanical arrangement that was used is shown in Fig. 1. The models were made from plastic strips $^1/_4$ x $^3/_4$ in. in cross section and 12 in. long, with 20-mil slots about 2 in. long milled through their centers. The strips were first loaded axially in compression, the compressive loads A, B and C being adjusted to make the photoelastic stress pattern uniform. Anti-buckling pins were put in light contact with the models at the places shown in the figure. After the compressive forces had been adjusted, small steel wedges were pushed into the milled slots from both sides, simultaneously, until cracks ran from each end of the slots part way down the models.

The purpose of the compressive stress was to suppress the components of tensile stress that would otherwise lie parallel to the crack surfaces when the wedges were pushed into the slots. When circumstances were good, the cracks ran straight down the models and remained parallel to the sides of the models. Although the compressive forces were effective in preventing transverse cracking, the planes of the cracks sometimes tended to twist about the longitudinal axis. No effective means of preventing the twisting was found; so, a number of cracks were made and the best one was selected for detailed study.

After the crack was made, the wedges were released, and the compressive forces were removed. Then light forces were reapplied to the wedges to produce stresses about the crack without causing it to run. The resulting stress pattern is shown in Fig. 2. The complete distribution of stresses was obtained along a series of lines, spaced 50 mils apart, running from the centerline to the edge of the model, as shown in Fig. 3. Measurements of fringe order along each of the lines were made with a Babinet-Soleil compensator in conjunction with a photometer. Isoclinic parameters were also determined directly on the model with the aid of the photometer.

The isoclinic lines were uncommonly sharp, indicating that extraneous twisting or bending of the model was negligible. The shearing stress distribution along each line was then calculated in the usual manner. The distribution of normal stresses on lines I through IX was obtained with the *shear-difference method.*[2] In applying this method to a given line, the two adjacent lines were used as the necessary auxiliary lines. Each line thus served a dual purpose. Integration proceeded from the outer edge to the centerline.

The three components of stress were recorded at the points of a 9 x 15 matrix adjacent to the tip of the crack. The points of the matrix are shown in Figs. 4 and 5, and measured values are given in Table 1 for each point. The numerical values represent the measured stresses divided by the average shearing stress (τ_{zv}) that existed on line I. Thus they hold for any value of applied load. The actual load, as determined from the measured shearing

TABLE 1—EXPERIMENTAL VALUES OF STRESS COMPONENTS AT THE CROSSES IN FIGS. 4 AND 5*

	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
I	0	−0.03	−0.07	−0.09	−0.07	−0.03	0.01	0.04	0.06	0.08	0.09	0.09	0.06	0.04	0.01	0
	5.3	4.79	4.19	3.44	2.69	1.98	1.13	0.36	−0.50	−1.29	−2.08	−2.96	−3.79	−4.61	−5.45	−6.26
	0	0.10	0.45	0.80	1.13	1.36	1.51	1.60	1.64	1.59	1.44	1.20	0.94	0.66	0.34	0
II	0	−0.13	−0.11	−0.02	0.08	0.18	0.24	0.27	0.25	0.21	0.16	0.10	0.06	0.03	0.01	0
	5.95	5.38	4.82	4.10	3.30	2.25	1.45	0.48	−0.56	−1.53	−2.57	−3.62	−4.58	−5.46	−6.35	−7.15
	0	0.10	0.44	0.85	1.25	1.48	1.61	1.65	1.65	1.57	1.42	1.21	0.89	0.58	0.28	0
III	0	0.06	0.31	0.64	0.90	1.01	1.01	0.94	0.80	0.64	0.47	0.30	0.17	0.07	0.02	0
	6.75	6.50	6.10	5.30	4.10	2.90	1.69	0.56	−0.64	−1.69	−2.73	−3.78	−4.90	−5.95	−7.06	−8.20
	0	0.29	0.69	1.19	1.53	1.68	1.68	1.61	1.53	1.40	1.21	1.00	0.78	0.53	0.27	0
IV	0	1.65	2.38	2.62	2.56	2.34	2.06	1.73	1.41	1.08	0.78	0.52	0.30	0.14	0.03	0
	8.44	9.71	8.75	6.18	3.94	2.33	1.04	−0.16	−1.04	−2.00	−3.05	−4.10	−5.22	−6.50	−7.86	−9.40
	0	0.29	1.68	2.14	2.00	1.69	1.43	1.23	1.05	0.89	0.73	0.58	0.42	0.28	0.14	0
V	—	7.15	6.16	5.13	4.17	3.40	2.74	2.18	1.70	1.27	0.90	0.59	0.34	0.15	0.04	0
	—	10.38	4.10	2.82	1.85	0.96	0.24	−0.64	−1.53	−2.50	−3.54	−4.58	−5.63	−6.66	−8.03	−10.60
	—	4.40	2.53	1.46	0.93	0.63	0.44	0.30	0.19	0.12	0.07	0.02	0	0	0	0
VI	—	7.36	6.10	5.00	4.10	3.32	2.68	2.14	1.65	1.25	0.89	0.60	0.35	0.16	0.04	0
	—	11.10	6.26	3.86	2.47	1.29	0.32	−0.56	−1.37	−2.25	−3.06	−4.02	−4.98	−6.03	−7.23	−8.84
	—	−2.61	−2.48	−1.93	−1.46	−1.15	−0.97	−0.86	−0.77	−0.70	−0.64	−0.57	−0.45	−0.32	−0.17	0
VII	1.90	2.20	2.54	2.60	2.48	2.20	1.90	1.56	1.24	0.93	0.67	0.45	0.26	0.13	0.03	0
	8.60	8.03	6.75	4.90	3.30	1.93	0.88	−0.24	−1.21	−2.01	−2.98	−3.78	−4.74	−5.70	−6.75	−7.88
	0	−1.42	−2.16	−2.42	−2.38	−2.16	−1.94	−1.74	−1.57	−1.41	−1.25	−1.07	−0.86	−0.63	−0.34	0
VIII	0.38	0.47	0.66	0.85	0.98	1.03	0.99	0.88	0.75	0.59	0 42	0.28	0.16	0.07	0.02	0
	6.50	6.19	5.55	4.58	3.54	2.50	1.37	0.32	−0.64	−1.61	−2.57	−3.38	−4.35	−5.15	−6.19	−7.07
	0	−1.04	−1.68	−2.06	−2.28	−2.33	−2.30	−2.20	−2.03	−1.84	−1.62	−1.37	−1.11	−0.80	−0.42	0
IX	−0.59	−0.52	−0.38	−0.22	−0.06	0.06	0.14	0.19	0.20	0.19	0.16	0.12	0.06	0.04	0.01	0
	4.90	4.66	4.26	3.62	2.90	2.17	1.37	0.48	−0.40	−1.29	−2.09	−2.90	−3.70	−4.58	−5.22	−5.86
	0	−0.80	−1.37	−1.76	−2.01	−2.16	−2.24	−2.24	−2.16	−2.04	−1.86	−1.61	−1.29	−0.91	−0.48	0

Key: σ_{zz}, σ_{xx}, τ_{zx}.

*Values given are in units of the average shearing stress on line I.

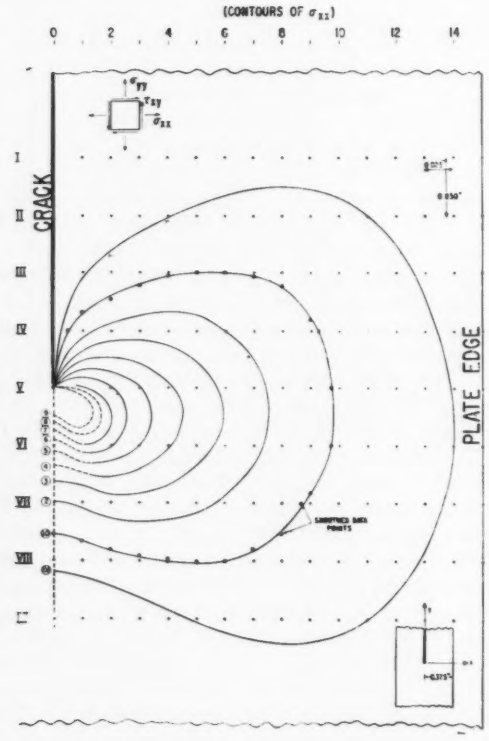

Fig. 4—Stress field near cleavage crack

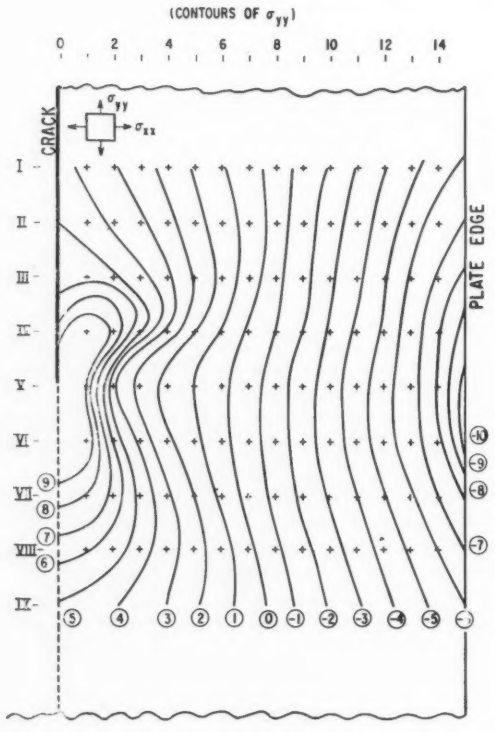

Fig. 5—Stress field near cleavage crack

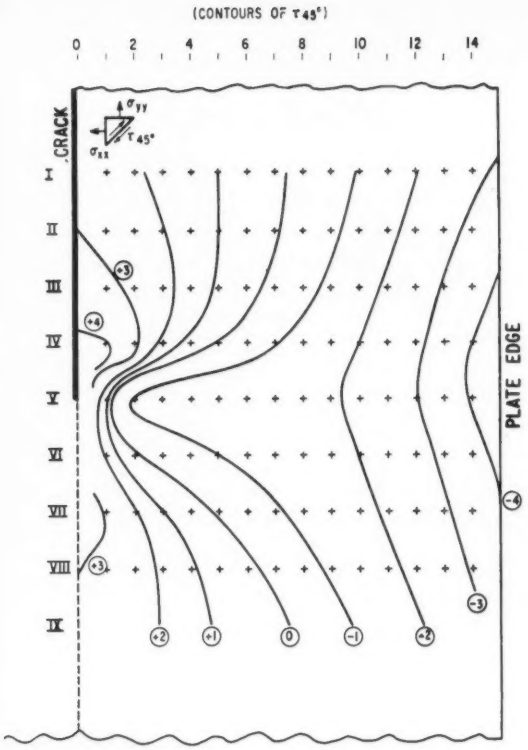

Fig. 6—Stress field near cleavage crack

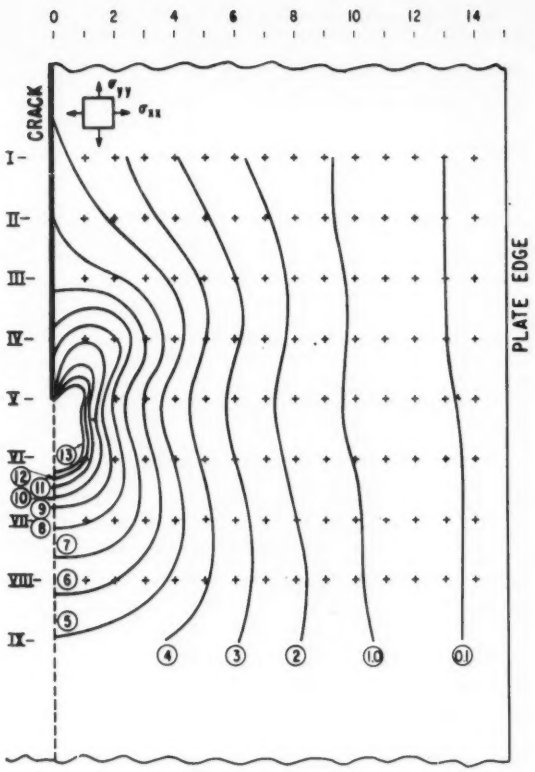

Fig. 7—Stress field near cleavage crack

force, was about 6.4 lb applied $1^1/_4$ in. from the crack tip.

From the measured values of σ_{xx}, σ_{yy}, the shearing stresses on planes making angles of 45 deg with the crack, $\tau_{45} = {}^1/_2(\sigma_{yy} - \sigma_{xx})$, were calculated and are plotted in Fig. 6. Also, values of the maximum tensile stress,

$$\sigma_{max} = \frac{\sigma_{xx} + \sigma_{yy}}{2} + \sqrt{{\tau_{45}}^2 + {\tau_{xy}}^2}$$

were calculated and are given in Fig. 7.

The Results

To give a basis for comparison, stress distributions for a crack in an infinite plate were calculated. For this purpose, the formulas given by Irwin[3] were used which are valid in the region near the tip of a crack (plane stress). The crack length is $2a$ (in this case taken to be 10 length units). Polar coordinates (r, θ) centered at $(0, a)$ are used. Then, if a uniform tensile stress, σ_0, is applied in the x-direction, the stresses are:

$$\sigma_{xx} = \sigma_0\sqrt{\frac{a}{2r}}\cos\left(\frac{\theta}{2}\right)\left[1 + \sin\left(\frac{\theta}{2}\right)\sin\left(\frac{3\theta}{2}\right)\right]$$

$$\sigma_{yy} = \sigma_0\sqrt{\frac{a}{2r}}\cos\left(\frac{\theta}{2}\right)\left[1 - \sin\left(\frac{\theta}{2}\right)\sin\left(\frac{3\theta}{2}\right)\right]$$

$$\tau_{yx} = \sigma_0\sqrt{\frac{a}{2r}}\cos\left(\frac{\theta}{2}\right)\sin\left(\frac{\theta}{2}\right)\cos\left(\frac{3\theta}{2}\right)$$

where $r \ll a$, and:

$$\tau_{45} = \frac{\sigma_{yy} - \sigma_{xx}}{2} = \sigma_0\sqrt{\frac{a}{2r}}\cos\left(\frac{\theta}{2}\right)\sin\left(\frac{\theta}{2}\right)\sin\left(\frac{3\theta}{2}\right)$$

$$\sigma_{max} = \frac{\sigma_{yy} + \sigma_{xx}}{2} + \sqrt{{\tau_{45}}^2 + {\tau_{xy}}^2} = \sigma_0\sqrt{\frac{a}{2r}}\cos\left(\frac{\theta}{2}\right)\left[1 + \sin\left(\frac{\theta}{2}\right)\right]$$

Contour lines of constant values of (σ_i/σ_0) are plotted in Fig. 8 for these various stresses.

It may be seen, by comparing Figs. 4 and 5 with Fig. 8, that the σ_{xx} and σ_{yy} contours are rather similar in shape. However, in the case of the cleavage model, the stresses decrease more rapidly with distance from the crack tip than they do for the case of the crack in the infinite plate. In Fig. 4, if one observes the variation of σ_{xx} along radial lines, it is found that it varies as $r^{-1/2}$ for small values of r, but then decreases much more rapidly as it must in order to reach zero before the free surface of the model is reached. Instead of being localized in a region of the order of the crack length in size, the stresses are localized to a region that is less than the half-width of the strip in size and is independent of the crack length.

The distributions of τ_{45} are quite different for the two cases as may be seen by comparing Fig. 6 with Fig. 8. Here the effect of the bending stresses in the cleavage model is quite apparent.

Figure 7 compared with Fig. 8 shows an important difference between the two distributions of the maximum tensile stress. The directions of the gradients are the important quantities. For the crack in an infinite plate (Fig. 8) the maximum gradient of the maximum tensile stress lies along the direction of the crack (y-axis). On the other hand, in the cleavage model, it may be seen that the maximum gradient (closest spacing of the contour lines) lies at a rather large angle to the plane of the cleavage crack. The angle is not sharply defined, but it is 70–80 deg measured from the direction of propagation.

A cleavage-type crack in an isotropic body tends to turn out of its own plane and run out the side of the specimen. This is in contrast to a crack in an infinite plate which tends to propagate in its own plane. In crystals, cleavage cracks tend to run straight, but this is the result of anisotropy in the strength (surface energy) of the crystal, and it occurs in spite of the unfavorable stress distribution. The influence of the stress distribution is shown by the fact that even crystals that have good cleavage (zinc, for example) often exhibit smoother cleavage surfaces when they are broken by a uniform tensile stress, than when the bending stresses of the cleavage-type specimen are present.

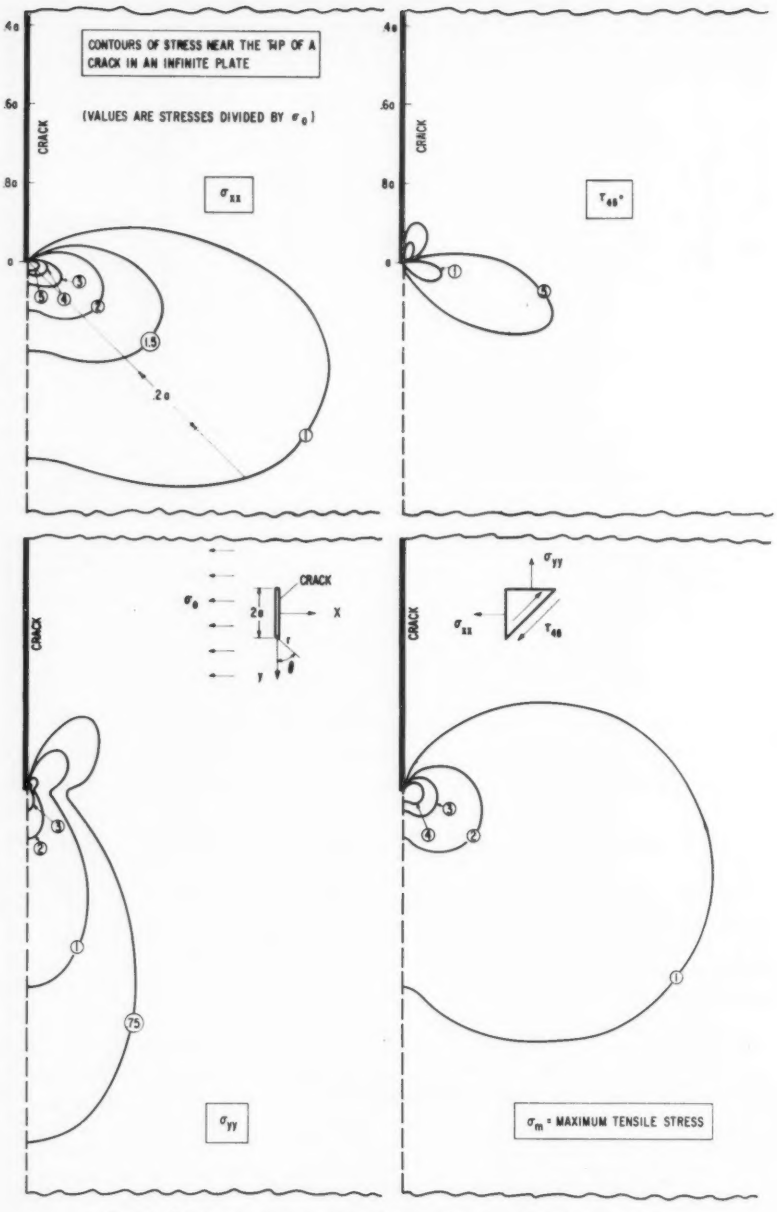
CONTOURS OF STRESS NEAR THE TIP OF A CRACK IN AN INFINITE PLATE
(VALUES ARE STRESSES DIVIDED BY σ_0)
CRACK
σ_{xx}
$\tau_{45°}$
σ_{yy}
σ_m = MAXIMUM TENSILE STRESS

Fig. 8—Contours of stress near the tip of a crack in an infinite plate

Closure

By means of photoelastic analysis, the stress distribution that is present near the tip of the crack when a body is split symmetrically with a wedge (called a "cleavage-type" specimen) has been determined. The distribution differs considerably from the case of a crack in an infinite plate. The region of high stresses is localized to a region of the order of the half-width of the body in size. For small distance, r, from the crack tip, the stresses vary as $r^{-1/2}$ just as in the infinite-plate case, but they decrease more rapidly at large r. The direction of the maximum gradient of the principal tensile stress makes an angle of 70–80 deg with the propagation direction of the crack, in contrast to the infinite-plate case where the maximum gradient lies in the plane of the crack.

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Model Analysis Method for Determining Buckling Load of Rectangular Frames

Plastic models used for experimental determination of the buckling loads of rigid-jointed rectangular frames when the buckling occurs within the elastic range

by H. P. Vaswani

SYNOPSIS—The paper deals with the experimental determination of the buckling loads of rigid-jointed rectangular frames when the buckling occurs within the elastic range. A setup for carrying out the experiments using models manufactured out of plexiglass is described. The critical load of a model having nonprismatic members is determined experimentally to demonstrate the simplicity of the experimental method and to emphasize its advantages over the theoretical method when dealing with complex problems of this type.

Introduction

The last few decades have witnessed a growing use of experimental stress analysis in practice. This is mainly due to the extreme simplicity of this method. Little extra labor, if any, is required with an increase in the complexity of the problem. The method is not limited by the usual refinements which must be made in theoretical analysis. For instance, the effect of axial thrust, shear deformation, varying moment of inertia and the peculiar stress distribution around corners, etc., is automatically taken into account. Intricate mathematical calculations are avoided. Often the model method provides the simplest and quickest method of arriving at a solution. This is particularly so when the setup designed for carrying out the experiment is simple and effective and a suitable material is chosen for the model.

Methods of analyzing structures with the aid of models fall into two categories.

The Indirect Methods[1–3]

These methods do not bear a direct relation to the manner of loading on the prototype. Influence lines for the stress elements (e.g., moment, shear or thrust) are plotted, from which the required reaction at any point under a given system of loads can be calculated. The indirect methods are different techniques of utilizing Maxwell—Betti's law of reciprocal deflections from which Müller-Breslau's principle of influence lines is derived. Influence lines are useful for linear structures, i.e., structures for which the law of superposition is valid. Buckling is a problem involving nonlinear structures. Hence, the use of influence lines in the experiments considering buckling is questionable.

The Direct Methods[4–6]

The model is loaded in the same manner as the prototype. The movements (rotations and translations) corresponding to the stress elements required are then measured directly by a suitable instrument, e.g., a microscope or a strain gage. The direct methods, being of special interest for the present paper, are discussed here in greater detail.

Southwell's Method

A widely known direct method of determining the critical load of struts experimentally is due to R. V. Southwell.[7] The model having the end conditions similar to those of the prototype is loaded and the deflections at a convenient point, say the middle of the column, are measured at various stages of loading.

The deflection curve of such a strut, Fig. 1, can be expressed in the form of a sine series,

$$v = \sum_{n=1}^{n=\infty} \frac{a_n}{\frac{P_{crn}}{P} - 1} \sin \frac{n\pi x}{L} \qquad (1)$$

where P_{crn} is the load required to cause the nth mode of buckling.*

Let $v = \delta$, at $x = L/2$. Then

$$\delta = \frac{a_1}{\frac{P_{cr1}}{P} - 1} + \frac{a_3}{\frac{P_{cr3}}{P} - 1} + \frac{a_5}{\frac{P_{cr5}}{P} - 1} \cdots \qquad (2)$$

When P approaches P_{cr1}, the first term of the series becomes predominant in relation to the rest and as a fair approximation we may write

$$\delta = \frac{a_1}{\frac{P_{cr1}}{P} - 1} \qquad (3)$$

This can be rewritten

$$\frac{\delta}{P} = \frac{1}{P_{cr1}}\,\delta + \frac{a_1}{P_{cr1}} \qquad (4)$$

an equation of a straight line plotted in Fig. 2. The inverse of the slope of this line is equal to the value of the critical load P_{cr1}. It is not necessary to

* *Timoshenko: Theory of Elastic Stability, 1936, p. 177.*

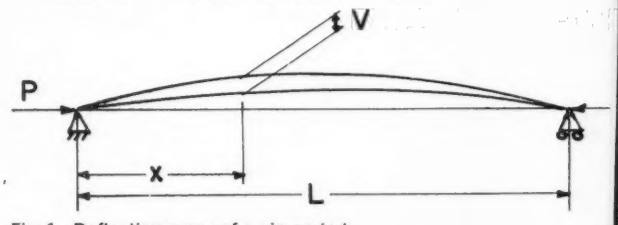

Fig. 1—Deflection curve of a pin-ended strut, length L, under an axial load, P

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load the model to its critical value. If the approximate critical value is known, the loading can be stopped before this value is attained, thus saving the model from destruction.

Southwell's method is particularly useful when the buckling load of an ideal column corresponding to a particular individual mode of failure is to be determined. For example, consider the case of the primary buckling load of an initially straight, centrally loaded, pin-ended column of uniform cross section. It is impossible to construct such a model in practice since, in reality, a column satisfying such ideal conditions does not exist. The effects of initial curvature, etc., influence the value of the buckling load of the model to such an extent that a fair approximation of the critical load is not obtained. However, if the experimental readings are plotted in the manner suggested by Southwell, it is possible to obtain a fair approximation to the theoretical critical load of the idealized column since the curve in this case is not affected by the final value of the critical load. One may say that Southwell's method predicts the buckling load of a perfect structure from an imperfect model.

Another advantage of Southwell's method, pointed out by Donnell,[8] is that by using harmonic analysis it is also possible to determine the values of the critical loads corresponding to the higher modes of failure. This very interesting application of Southwell's method has been used by Granholm[9] in determining buckling loads corresponding to different modes of failure of nailed timber columns.

Donnell discussed the use of Southwell's method and suggested another manner of plotting the straight line. Equation 3 rewritten as

$$P = -a_1 \frac{P}{\delta} + P_{cr1}$$

is replotted in Fig. 3. Measuring the slope of the straight line is thus avoided, since producing the straight line to cut the ordinate gives the value of P_{cr1} directly.

Lundquist[10] extended Southwell's method and applied it to framed structures. Lundquist suggested a stiffness criterion for structures prevented from sway. An arbitrarily small external moment M is applied to some joint j and the resulting joint rotations θ_j at increasing axial loads P are calculated. When θ_j is infinitely large, the frame is said to have zero stiffness and is thus unstable. Lundquist used the cross moment distribution to calculate the values of θ_j. Since the convergence, on which this method is based, deteriorates in the vicinity of the critical load and is absent at values higher than the critical load, it is necessary to arrive at the critical value by extrapolation. Lundquist used Southwell's method to extrapolate the value of the critical load. According to Lundquist's method, the curve, obtained by plotting the values of θ_j as abscissa against those of θ_j/P as ordinates, yields a straight line the inverse slope of which is equal to the value of the primary critical load, Fig. 4.

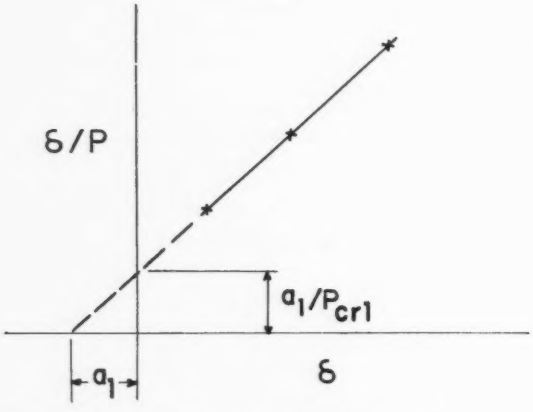

Fig. 2—Relation between deflection at mid point, δ, and δ/P, when the axial load, P, approaches the primary buckling load, P_{cr1}

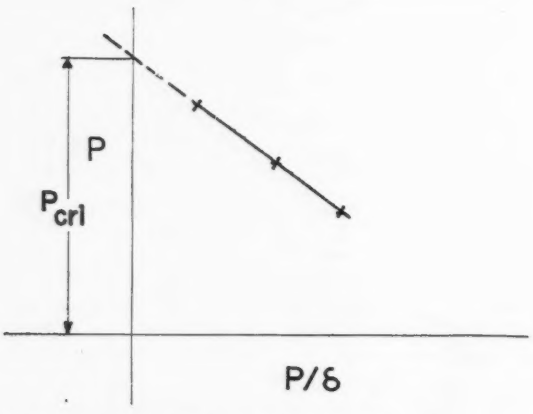

Fig. 3—Relation between axial load, P, and P/δ when P approaches the primary buckling load, P_{cr1}

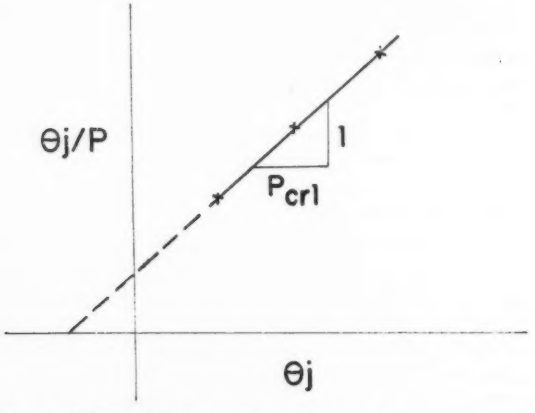

Fig. 4—Relation between joint rotation, θ_j, and θ_j/P when the axial load, P, approaches the primary buckling load, P_{cr1}

Southwell's method has been used extensively with success. Donnell[8] extended its use to beams on elastic foundations, plates supported on three and four sides, and cylinders under axial compression, and discussed its use on struts in the plastic range. Donnell also showed that this method can be expected to give reasonable results if applied to cases in which buckling does not introduce appreciable second-order stresses, as when a developable surface buckles into a nondevelopable one. As already mentioned, Granholm[9] applied Southwell's method to built-up columns and Lundquist[10] extended its use to framed structures. Hoff[11] and others applied it to columns subjected to rapid compression loads.

However, Southwell's method is subjected to limitations. As Donnell has pointed out, and as can be seen from the previous equation, the assumption holds good only in the vicinity of the primary critical load. For values smaller than the critical load (how small depends upon the slenderness of the structure), the buckling effect is not predominant and the curve is not straight. At the other extreme, when P approaches P_{cr1} the deflections could be so large that due to combined bending and direct action, yielding may occur, as shown in Fig. 5, again deviating the curve from a straight line. The success of the method thus depends upon there being an identifiable straight portion between these limits. Southwell gave a proof and showed that such a straight portion exists for a slender pin-ended strut. However, for certain structures, the curve may not have a straight portion at all. Then the readings obtained from the experiment might be thought to have a wide scatter and a straight line interpolated as shown in Fig. 6. This would give a completely misleading idea of the buckling load.

As discussed above, Southwell's method determines buckling loads corresponding to individual modes of failure. In a structure met with in practice, where, in addition to the primary buckling mode, other buckling modes also are present, the value of the primary buckling load alone, obtained by Southwell's method, may have no relation to the value of the actual buckling load. In such cases, it is preferable to consider the stiffness of the structure as a whole and plot the readings obtained from the experiment in the manner discussed in the following section.

Stiffness as a Function of Axial Loads

For rigid-frame structures, it is not necessary to follow Southwell's method or its modifications. Instead of this, a conception of the stiffness as a function of the axial loads is introduced. An arbitrarily small moment M is applied at a convenient joint as suggested by Lundquist.[10] Maintaining the value of M constant throughout the experiment, the rotations θ_j at a convenient joint j, corresponding to the increasing axial loads P are recorded. M/θ_j then represents the stiffness of the structure, which progressively reduces with the increasing axial loads P. At the critical load, M/θ_j reduces to zero. If a curve is plotted with M/θ_j as the ordinate against P as the abscissa, the value of the loads P at which the curve cuts the abscissa is the critical load (see Fig. 7). This is equally true for structures prevented from sway and for structures subjected to sway. In the latter case, however, it is more practical to replace the arbitrarily small moment M by an arbitrarily small transverse load Q and to study the resulting deflection δ at any convenient point. Q/δ is then the measure of stiffness; its value reduces to zero at the critical load (see Fig. 7).

The notion of stiffness is commonly used for isolated members in the moment distribution and the

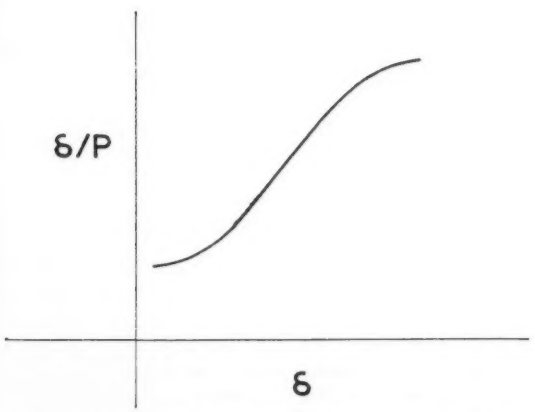

Fig. 5—Relation for a pin-ended strut between deflection at mid point, δ, and δ/P when the axial load, P, varies from a small value to the primary buckling load, P_{cr1}. The curve has a definite straight portion

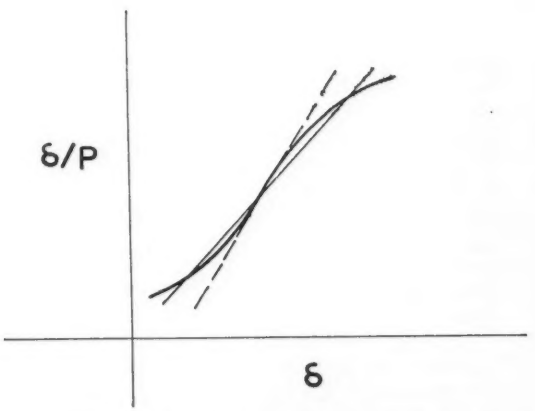

Fig. 6—Relation for a strut, not necessarily pin ended, between deflection at mid point, δ, and δ/P when the axial load, P, varies from a small value to the primary buckling load, P_{cr1}. The curve has no definite straight portion

slope-deflection methods. The values of stiffness for members subjected to axial loads have been tabulated by Lundquist and Kroll,[12] Hansbo,[13] Winter[14] and others. In this case, the stiffness is defined as the moment required to cause a rotation of unit angle at a simply supported end of the given member. If the values of stiffness are plotted as ordinates with the values of axial loads as abscissa, a curve similar to that of Fig. 8 is obtained from which the value of the critical load can be determined.

The notion of stiffness has also been used by Galletly and Reynolds,[15] Livesely[16] and Merchant.[17]

It is of considerable advantage to regard the stiffness as a function of axial loads and to study the stiffness of the frame as a whole. The curve of stiffness not merely determines the value of the critical load, but also supplies some vital and useful information in addition. It shows a clear picture of the situation corresponding to a given value of the axial loads.

The stiffness curve also helps to give a clearer definition of buckling. The critical load of a bar is defined as the axial load at which the slight lateral deflections produced by a slight lateral load do not vanish when the lateral load is removed.† According to the stiffness curve, buckling occurs when the stiffness reduces to zero. While the conventional definition suggests an instantaneous change from the unstable stage, the notion of stiffness shows the gradual decrease of stiffness before the structure finally passes into the unstable stage.

The stiffness curve also shows that special treatment is required when designing a structure to withstand axial loads. Figure 8 shows δ - P curves of two structures, A and B, whose final and initial stiffnesses are the same. The stiffness corresponding to the working load lies somewhere in between, depending upon the factor of safety allowed with regard to buckling. From the figure it is easily seen that the Structure B is inferior to the Structure A, since a slight increase in the axial load causes a much greater deflection of the former structure than of the latter.

† *Timoshenko: Theory of Elastic Stability, 1936, p. 65.*

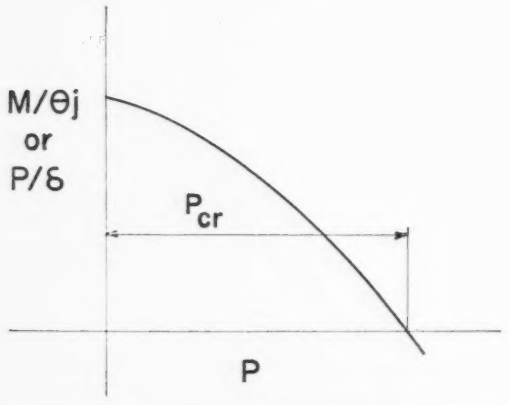

Fig. 7—Relation, between the stiffness, M/θ_j, of a structure free to sway and axial load, P, under a small and constant moment, M, when P varies from a small value to P_{cr}. For a structure prevented from sway, the stiffness is indicated by Q/δ where Q is a small and constant transverse load and δ is deflection at a given convenient point

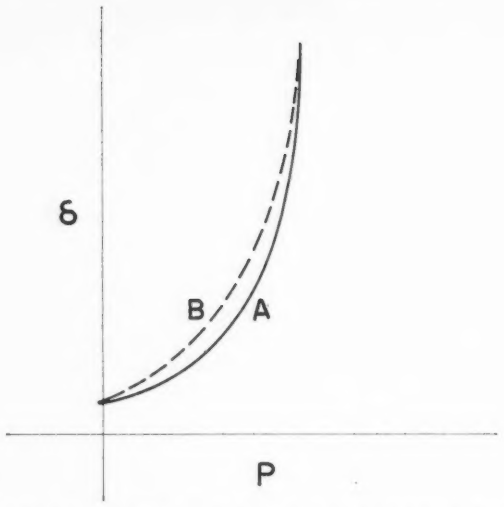

Fig. 8—Relation between deflection, δ, and axial load, P, of two structures, A and B, whose initial and final stiffnesses are identical, but intermediate stiffnesses are different. The figure clearly indicates the inferiority of structure B since a given increase in axial load causes a much greater deflection of the structure B

It may be of interest to consider the significance of the negative portion of the stiffness curve under compressive axial loads. The negative stiffness also appears in the tables of stiffness[12–14] for the isolated members discussed above. In this case, the value of the moment required to maintain a unit angular rotation at the support decreases with increasing axial loads P, so that at the primary critical load no moment is necessary to maintain the unit rotation. When the axial load increases above the primary critical load, a moment in the opposite direction is required to maintain the same unit rotation, thus yielding the negative stiffness.

A method of determining the buckling load of a model using the stiffness curve discussed above will now be described.

Selection of the Material for the Model

Experiments considering the stability of structures have been made on models manufactured out of traditional materials, e.g., timber and steel. In recent years, plastic has been introduced as a material for the model. Plastic can be sawed and machined with ease and welded together by coating the contact surfaces with a solvent-type cement. It has a low modulus of elasticity and can withstand relatively large displacements. This makes it a material of special value for experiments considering buckling

of structures. While plastic models have been widely used in experiments disregarding the effects of axial loads, their use, in experiments on the stability of structures, as far as it can be ascertained by the author, has not been reported previously in engineering literature. Hence, it was thought that a paper, furnishing evidence that plastic can be successfully used for such experiments, may not be without value since the use of plastic considerably simplifies the procedure of the experiment.

Unfortunately plastic also possesses certain other characteristics, namely a fairly high coefficient of expansion and the creep, which render it undesirable for use in the direct methods. The errors due to the high coefficient of expansion are negligibly small under conditions normally prevailing in the laboratory and can be disregarded. The errors due to the creep, on the other hand, may have an appreciable effect on the accuracy of the final results. Several methods have been devised for analyzing stable structures by direct methods using plastic models.[2-5] Special devices are used to circumvent the effect due to the creep of the material. Although the error due to creep can be serious, it is possible to reduce its range to within narrow limits, as borne out by the following discussion.

The results of a creep test on a beam manufactured out of plexiglass are shown in Fig. 9. From this it is seen that the rate of variation of the modulus of elasticity E decreases rapidly during the first few minutes. The variation, of the modulus of elasticity in the interval 5 to 10 min after loading, is as little as 2.8%. A period of 5 min is sufficient to permit the taking of the required readings corresponding to a particular loading. If an average value of E = 32,900 kg/cm^2 is used, the possible error from this source is of the order of $\pm 1^1/_2\%$. An error of this order can be reasonably tolerated in the experiments under discussion. Hence, if readings are taken in the interval suggested above, plastic models can be used for direct methods without resort to special devices and without undue loss of accuracy. This approach closely resembles that used by Sparkes[6] to circumvent the effect of creep of plastic. He suggested taking instantaneous readings 3 min after loading and used the "three-minute modulus."

Using the stipulations discussed above, the value of the buckling load can be determined in the following manner. A model of the prototype is manufactured out of plexiglass. An arbitrarily small transverse load Q is applied at a convenient point on the model and its value maintained constant throughout the experiment. Transverse displacements δ, at any convenient joint of the structure, are measured under the increasing multiples of the axial loads P. The values of the ratio Q/δ are plotted as ordinates against the corresponding values of the axial loads P as abscissa. The curve thus obtained yields the value of the critical load, as explained in Fig. 7. In practice, loading is stopped shortly before the curve cuts the abscissa, thus saving the model from destruction. The critical load is obtained by extrapolating the curve so that it cuts the abscissa.

The results of a series of experiments using a specially designed setup will now be described.

The Model and the Experimental Procedure

Model

A model of the frame is fabricated out of a plexiglass (methyl methacrylate) sheet (see Fig. 10). The frame is provided with footings. By varying the number and the position of the pins introduced in the footings, either of the columns can be hinged or fixed at the base as required.

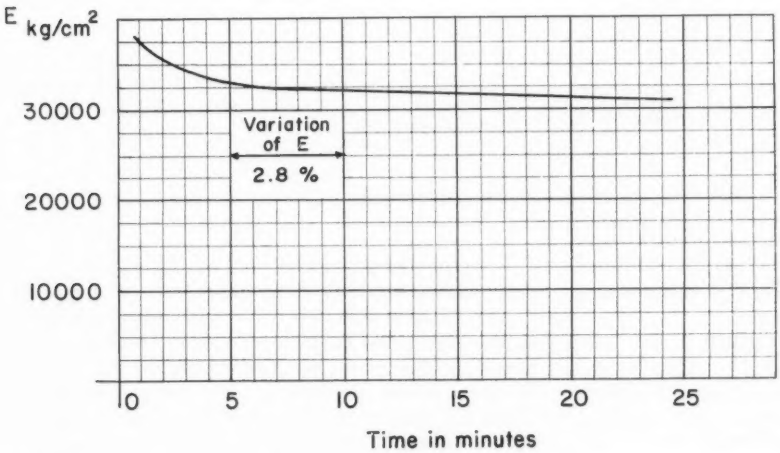

Fig. 9—Variation of modulus of elasticity, E, with time, of plexiglass showing decreasing rate of variation of E as the time interval after application of load increases

Mounting the Model

Buckling of the model out of its own plane, also called warping, must be carefully guarded against. This is best attained by sandwiching the model in such a manner that free movements of the model in its own plane are possible. This is one of the reasons why the model is mounted in a horizontal position. The model (3) rests on plexiglass packings (2) which in turn rest on SKF 5-mm diam balls rolling freely in ball races (1) manufactured out of plexiglass. Brass strips (4) are placed on the top of the

model. This arrangement provides the desired setup permitting nearly friction-free movements in the plane of the model and simultaneously preventing the model from buckling out of its own plane.

Loading

The loading device also presents special problems. It is necessary for the loading line to follow the movement of the model so that the applied axial loads maintain their direction despite the movement of the model. This is achieved by manufacturing a movable edge bearing (7) resting on SKF 5-mm balls rolling on a brass plate. This arrangement permits smooth movement of the edge bearing. After every increment of axial loads, the position of the edge bearing is adjusted by an amount corresponding to the movement of the model. The axial loads P are applied by weights hung at one end of a 1-mm piano wire (6) passing over a ball bearing (8) welded on to the movable edge bearing (7). The other end of the piano wire is attached to the loading beam (5) manufactured out of a 6-mm aluminum strip, the edges of which are bent and grooved to engage in the model as shown.

The arbitrarily small load Q is applied through 0.2-mm piano wire (9) passing over a pulley (10).

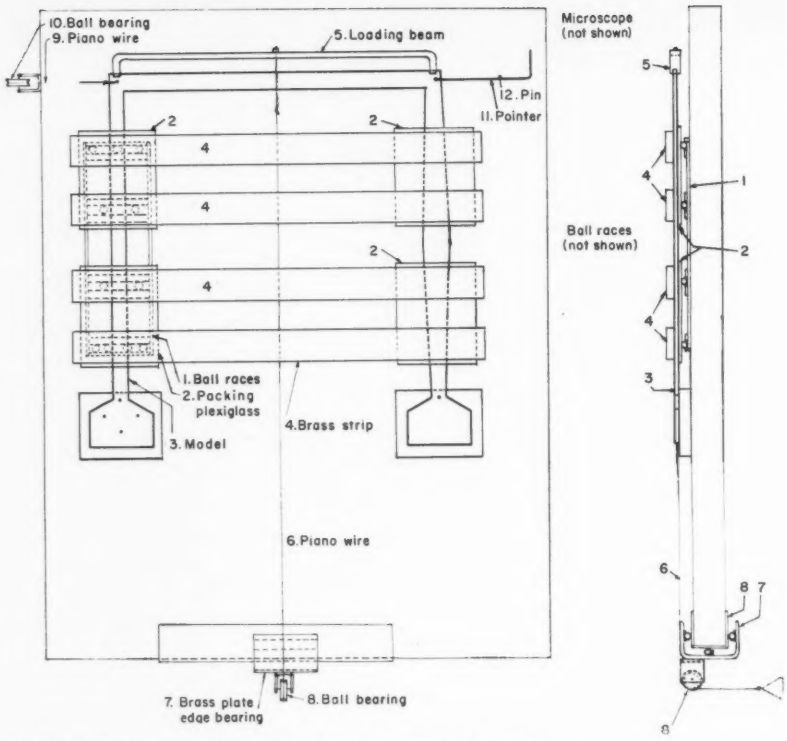

Fig. 10—Setup for the experiment showing a plexiglass model sandwiched between brass strips and packings, supported on ball races, to allow free movement without warping. The figure also shows the loading arrangement

Readings

The movement of the pointer (11) gives the value of the displacement δ. The pointer is designed to rotate freely in the joint of the frame and, thus, is not affected by the rotation of the joint. A pin (12) is fixed in position so that the centerline of the pointer, when the pointer is in contact with the pin, always lies along the straight line produced through the joints of the frame. When taking readings, care is taken to see that the pointer (11) is in contact with the pin (12).

In the present series of experiments, a Zeiss traveling microscope was used to measure the displacements. This was the only expensive piece of equipment used and measured readings to a fine degree of accuracy. By incorporating a rack and pinion device between the pointer and the pin it is possible to magnify the small displacements so as to dispense with the use of the microscope. Alternatively, where the displacements are appreciably large, it may be possible to obtain reasonable results by using a vernier scale or an ordinary scale and a magnifying glass.

The Scale of the Model

If the scales of the model (M) are related to those of the prototype (P) such that the

length	$L_p = n_1 L_M$
modulus of rigidity	$E_p = n_2 E_M$
moment of inertia	$I_p = n_3 I_M$
axial load	$P_p = n_4 P_M$
transverse load	$Q_p = n_5 Q_M$

and if the scales are so chosen that $n_4 = n_2 n_3 / n_1^2 = n_5$, then as a fair approximation,

$$\delta_p = n_1 \delta_M \qquad (5)$$

For the derivation of the above equation, see Appendix at the end of the paper.

The relation between δ_p and δ_M given above, requires that only the flexural rigidity at every cross section of the model need be correspondingly similar to that of the prototype. The error, introduced when the cross-sectional areas of the prototype and of the model are not correspondingly similar, is neglected. The approximation involved in this assumption is referred later on, under subheading "Direct Stress."

Examples

The accuracy of the method just presented will now be demonstrated by application to single-story, single-bay symmetrical rectangular frames subjected to axial loads. The critical load of these frames can be easily calculated theoretically and the final results conveniently checked for comparison. The behavior of the frame at loads other than the critical can be determined theoretically by the methods given by Merchant,[17] Bolton[18] or the author.[19] Using these methods, the theoretical stiffness curve is drawn for every example given below thus offering a check on the stiffness curve determined by the experimental method.

Example 1

The critical load of the rectangular portal frame, Fig. 11, having columns fixed at the base, and having members of constant flexural rigidity, will be determined experimentally.

A model of the frame is fabricated out of a 3-mm plexiglass sheet. Using the setup shown in Fig. 10, the values of the deflections at different loads are measured. The value of the transverse load Q used in this experiment was 0.05 kg. The curve of stiffness is shown in Fig. 12 which yields the value of the critical load equal to 16.6 kg. The corresponding value obtained theoretically by the method given by Bleich[20] (p. 254) is equal to $2(0.923\ \pi)^2\ EI/L^2 = 17.7$ kg, which deviates by 7% from the result obtained experimentally.

Example 2

The critical load of the rectangular portal frame, Fig. 11, having columns pinned at the base, and having members of constant flexural rigidity will be determined experimentally.

The same model, used in Example 1, is used for this example also after making the necessary adjustments at the footings of the columns. The values of the displacements for a value of the transverse load Q equal to 0.05 kg. are measured. The curve of stiffness is shown in Fig. 12 which yields the value of the critical load equal to 4.25 kg. The corresponding value obtained theoretically by the method given by Bleich[20] (p. 253) is equal to $2(0.461\ \pi)^2\ EI/L^2 = 4.42$ kg. which deviates by 4% from the result obtained analytically.

Example 3

The critical load of the unsymmetrical rectangular

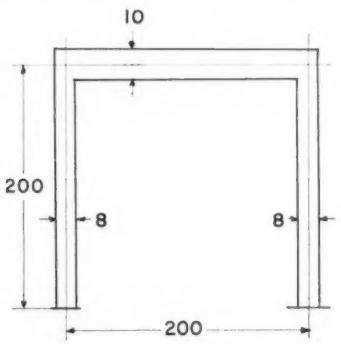

Fig. 11—Dimensions of model of a rigid-jointed rectangular frame having members of constant bending rigidity, cut out of 3-mm plexiglass sheet. All dimensions are in mm

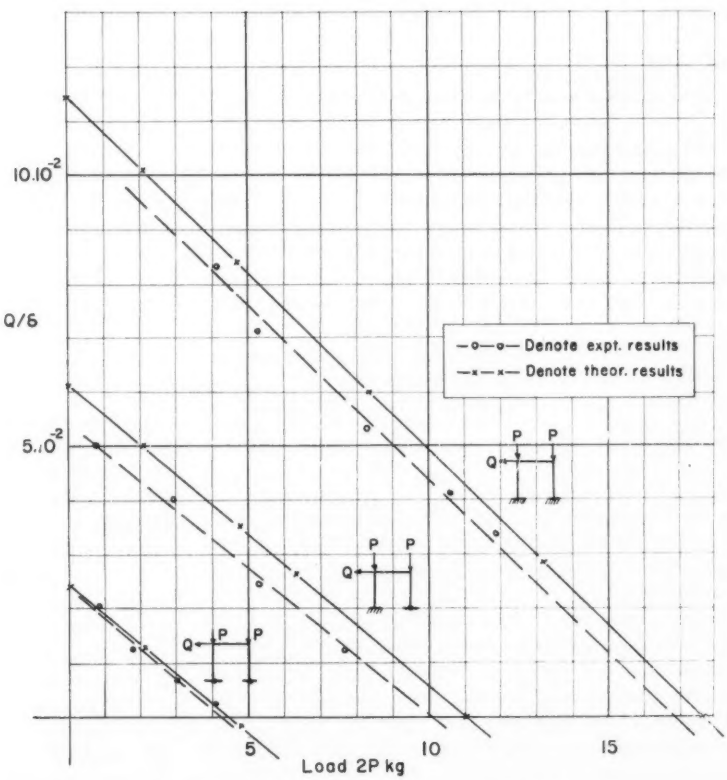

Fig. 12—Stiffness curves, theoretical and experimental, for the model shown in Fig. 11, corresponding to different end conditions; i.e., (a) both ends pinned, (b) one end pinned and the other fixed, (c) both ends fixed

portal frame, Fig. 11, having one column fixed and the other pinned at the base, will be determined experimentally.

The same model used in Examples 1 and 2 is used for this example also after making the necessary adjustments at the footings of the columns. The values of the displacements for a value of the transverse load Q equal to 0.05 kg are measured. The curve of stiffness is shown in Fig. 12 which yields the value of the critical load equal to 10 kg. The corresponding value obtained theoretically by the method given by Hansbo[13] is equal to 10.8 kg which deviates by 8% from the result obtained analytically.

The advantage of the experimental method over the theoretical method is particularly shown when dealing with structures having members of varying flexural rigidity. The critical loads of such frames is difficult to calculate theoretically for the lack of availability of the beam constants required in the theoretical analysis. The author[21] has developed a numerical method for the calculation of the beam constants for nonprismatic members under axial loads and applied it to the case of a member having its depth varying uniformly from the supports to the center of the span. Using these constants, the stiffness curves of the frame, Fig. 13, have been determined theoretically to offer a check on the stiffness curves obtained by the experimental method.

Example 4

The critical load of the rectangular portal frame, Fig. 13, having columns pinned at the base and having one member of varying bending rigidity will be determined experimentally.

A model of the frame is fabricated out of a 3-mm plexiglass sheet. Using the setup shown in Fig. 10, the values of the deflections at different loads are measured. The value of the transverse load Q used in this experiment was 0.05 kg. The curve of stiffness is shown in Fig. 14 which yields the value of the critical load equal to 5.6 kg. The corresponding value obtained theoretically by the author[19] is 6.1 kg which deviates by 9% from the result obtained experimentally.

Example 5

The critical load of the rectangular portal frame, Fig. 13, having its narrower column fixed at the base and the other pinned, will be determined experimentally.

The same model used in Example 4 is used for this example also after making the necessary adjustments at the footings of the columns. The values of the displacements for a value of the transverse load Q equal to 0.05 kg, are measured. The curve of stiffness, shown in Fig. 14, yields the value of the critical load equal to 12.8 kg. The corresponding value obtained theoretically by the author is equal to 13.7 kg, which deviates by 7% from the results obtained experimentally.

Sources of Approximation in the Experiment

Creep and Coefficient of Expansion

The range of possible error introduced by the creep and the varying coefficient of expansion of the material used for the model has already been discussed.

Plastic Sheet

The thickness of the plastic sheets commercially available may vary, within certain limits, from the nominal thickness claimed by the manufacturers. It is a good practice to measure the thickness of every sheet individually and use this value in calculating the flexural rigidity at a cross section of the model. The variation in one and the same sheet, as a rule, is not appreciably large.

Friction

The approximation introduced by friction between plastic and balls is negligibly small. Friction makes the stiffness appear higher than its real value. This can be partly compensated by allowing only negative tolerances. Permissible tolerances obviously depend upon the dimensions of the model. In the present examples the value of the tolerance aimed at was ± 0.08 mm. The maximum variation of the moment of inertia of the narrowest member was thus limited to $\pm 3\%$.

Transverse Load Q

The transverse load Q introduces extra moments and axial loads in the members. If the value of Q is chosen judiciously, the error due to this source can be made reasonably small.

In order to save the model from destruction, loading is discontinued before the stiffness is reduced

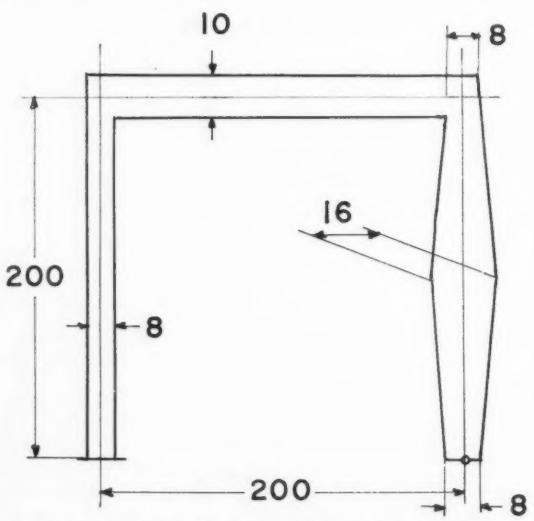

Fig. 13—Dimensions of model of a rigid-jointed rectangular frame having one member of varying bending rigidity, cut out of 3-mm plexiglass sheet. All dimensions are in mm

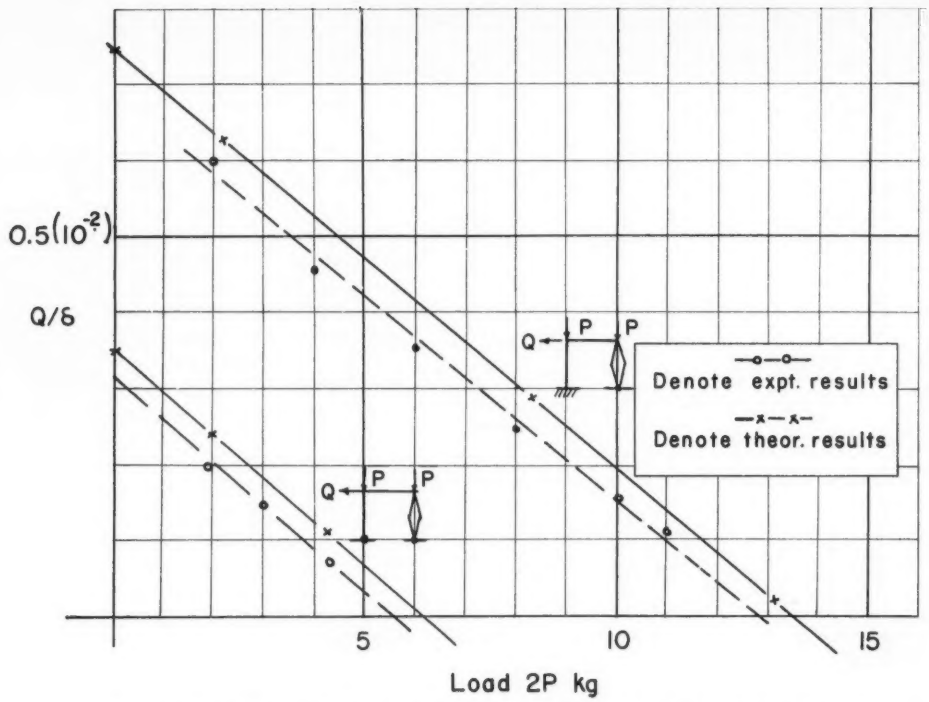

Fig. 14—Stiffness curves, theoretical and experimental, for the model shown in Fig. 13, corresponding to different end conditions; i.e., (a) both ends pinned, (b) narrower column fixed at the base and other end pinned

to zero. The value of the critical load is obtained by extending the stiffness curve to cut the abscissa, as explained above. To reduce the error due to extrapolation, it is desirable to continue the loading to a point as close to the point of zero stiffness as practicable. The values of the deflection increase rapidly as the point of zero stiffness is approaced. The value of the transverse load Q should be so chosen that the model can withstand the high deflections required to arrive at a point reasonably close to the point of zero stiffness.

Direct Stress

The model automatically takes the effect of the direct stress into consideration. Since this effect is normally neglected in the theoretical methods, the values of stiffness obtained by a conventional theoretical method are higher than the corresponding values obtained by the model method, the latter being nearer to the true values.

However the values obtained by the model method may not correspond to those obtained by the "exact" methods which do take the effect of the direct stress into account, if the cross-sectional areas of the model are not correspondingly similar to those of the prototype. For a given model, the difference in the values obtained by the two methods will increase with increasing values of the direct stress. On the other hand, an increment in the direct stress is coupled with a reduction in the stiffness. Since the direct stress decreases in importance in the region of low stiffness, the error due to this effect usually will be small. Hence, as a fair approximation, it is sufficient to design the model so that only the flexural rigidity at every cross section of the model is correspondingly similar to that of the prototype. Equation 5, previously given, is based upon this approximation, the model being designed so that the area at every cross section of the model need not be correspondingly similar to that of the prototype.

Conclusions

Plexiglass can be used as a material for models in experiments considering the effect of axial loads. Reasonable results are obtained by using the setup for carrying out the experiments developed above.

The worth of the experimental method is particularly shown when structures to be dealt with have members of varying flexural rigidity or are subject to other complexities which render the theoretical treatment too intricate, if not intractable. The experimental analysis then offers the simplest solution within a reasonable degree of accuracy.

In a structure subjected to small horizontal loads, e.g., due to wind, a small increase in axial loads may produce disproportionately large deflections, rendering the otherwise stable structure unsuitable for practical use. This emphasizes the desirability of calculating the behavior of a structure under increasing axial loads. The method developed above then assumes special significance.

Acknowledgments

The author is indebted to S. O. Asplund of Chalmer's University of Technology, for encouragement to carry out the experiments. The author owes a debt of gratitude to A. J. S. Pippard and Z. Makowski of the Imperial College of Science and Technology who first initiated him to the use of models in structural analysis. Acknowledgment is also due to S. Hansbo for his valuable criticism.

APPENDIX

Structural Similitude

In the following, application of the principles of similitude governing the relationships between a model and its prototype, to the case where in addition to transverse loads axial loads are also present, is discussed.

Consider the case of a simply supported strut, having uniform flexural rigidity under the action of a transverse load Q (see Fig. 15).

The equation of the deflection curve of such a strut is given by Timoshenko.‡ The expression for the deflection at the center of the span is

$$\delta = \frac{Q}{2P\sqrt{P/EI}}\left[\tan\sqrt{\frac{PL^2}{4EI}} - \sqrt{\frac{PL^2}{4EI}}\right]$$

A model M is made of this strut such that the following relations hold:

length	$L_p = n_1 L_M$
modulus of elasticity	$E_p = n_2 E_M$
moment of inertia	$I_p = n_3 I_M$
axial load	$P_p = n_4 P_M$
transverse load	$Q_p = n_5 Q_M$

The expressions for the deflection of the model and the prototype can be written

$$\delta_M = \frac{Q_M}{2P_M\sqrt{P_M/E_M I_M}}\left[\tan\sqrt{\frac{P_M L_M^2}{4E_M I_M}} - \sqrt{\frac{P_M L_M^2}{4E_M I_M}}\right]$$

$$\delta_p = \frac{Q_p}{2P_p\sqrt{P_p/E_p I_p}}\left[\tan\sqrt{\frac{P_p L_p^2}{4E_p I_p}} - \sqrt{\frac{P_p L_p^2}{4E_p I_p}}\right]$$

If the scales are so chosen that $n_4 = n_2 n_3/n_1^2$, the expression for the deflection of the prototype reduces to $\delta_p = n_1 n_5 \delta_M/n_4$ and if, in addition $n_4 = n_5$, $\delta_p = n_1 \delta_M$.

Hence, if the scales of the loads are chosen properly, the principle of structural similitude can be applied to structures subjected to axial loads. In this analysis the effect of the axial strains is disregarded as explained under the subheading "Direct Stress" of this paper.

If similar expressions for δ were available for structures more complicated than the simple strut, similar results could be obtained correspondingly. Unfortunately, it is not easy to obtain such simple expressions for the type of structures which form the subject of the paper. On the other hand, a corollary between the case of the simple strut and that of a more complicated frame appears reasonable. Though a formal rigorous proof is lacking, it seems reasonable to accept the proposition that structural similitude applies even for the more complicated structures.

It may not be out of place here to remind ourselves that this acceptance of a proposition in good faith has a close parallel in a similar problem discussed under the heading "Southwell's Method." Lundquist, Donnel, Merchant and others have extended eq 3 to structures other than the simple strut even though the formal proof for such structures is still lacking.

‡ *Timoshenko, Theory of Elastic Stability, 1936, p. 4.*

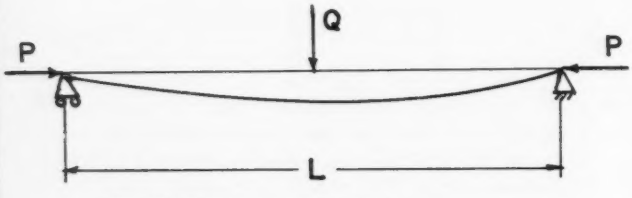

Fig. 15—Simply supported strut

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Measurement of Yield Points of High Polymers with Resistance-wire Dynamometer

Dynamometer utilizing electric-resistance wire can be used to measure dynamic or transient changes in load

by Katsuhiko Ito

ABSTRACT—A new load dynamometer utilizing electric-resistance wire consists of a cylinder around which the resistance wire is coiled tightly. When the cylinder is compressed axially and deforms elastically, the resistance wire is strained an amount equal to the circumferential strain of the cylinder resulting from Poisson's ratio. The dynamometer is designed with attention to electrical insulation, rate of heat dissipation, compensation for temperature change of electric resistance and material for the stressed body. The reading of this load dynamometer is not affected by the bending moment due to eccentric loading, and the bridge unbalance to be measured is larger than by any other type of strain gage utilizing resistance wire.

This load dynamometer can be applied to the measuring of dynamic or transient changes in load. As one example, the tardy yielding phenomenon of thermosetting cast resins has successfully and reasonably been measured by this dynamometer without any inertia effect.

Introduction

The measuring of dynamic or transient changes in load is often necessary in engineering problems, but cannot be done accurately by most conventional testing machines. Various methods[1, 2] have been tried. The load dynamometer utilizing a piezoelectric crystal is one of the most outstanding methods for this purpose. It has been used successfully in many engineering problems.[3–7] Recently, the electric-resistance wire gage[8] has been applied to a wide variety of engineering problems. However, careful handling, special environment and advanced techniques and instrumentation are necessary for the use of both a piezo dynamometer and the conventional electric strain gage.

In this paper a load dynamometer utilizing electric-resistance wire, having better stability and being easier to handle, is described and its application to evaluating the yield points of high-polymer solids is given.

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Principle and Merit

The stressed body in this load dynamometer is a cylinder, either solid or hollow, around which electric-resistance wire is tightly coiled with small pretension, as shown in Fig. 1 (a). The load to be measured is usually compression. When the cylinder is compressed axially, it shortens but the resistance wire is stretched because of the lateral deformation of the cylinder, in accordance with Poisson's ratio of a stressed body, as shown in Fig. 1 (b). Therefore, the axial compressive load can be measured by evaluating the change of electric resistance of the wire.

Because Poisson's ratio of common metals is approximately one third, the circumferential strain of the cylinder (i.e., the strain of resistance wire) is smaller than the axial compressive strain of the cylinder. This is an undesirable feature of this load dynamometer. Nevertheless, it has the following merits:

(*a*) Dynamic or transient changes in load can be measured without being influenced by inertia or hysteresis because of the very high natural fre-

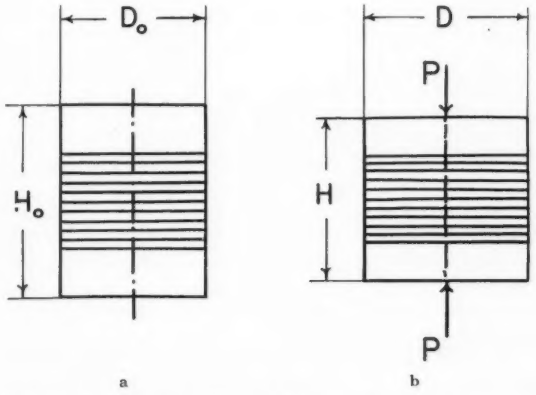

Fig. 1—Fundamental principle of load dynamometer

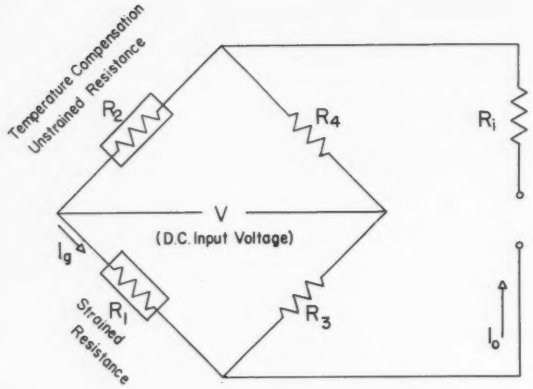

Fig. 2—Temperature-compensation electric circuit

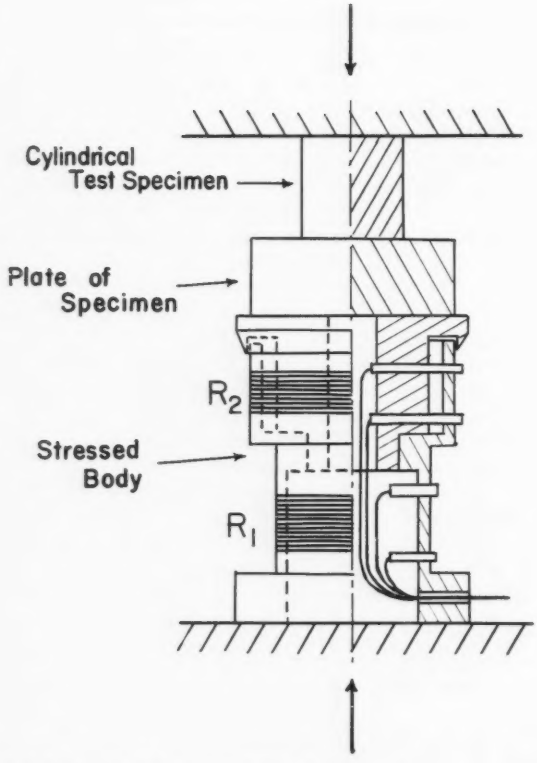

Fig. 3—One type of stressed body being used in compression test of cylindrical specimen

quency resulting from the very large spring constant of the cylinder and the linearity of its stress-strain relation.

(*b*) Compensation is made for the eccentricity or bending effect in loading.

(*c*) The magnitude of electrical resistance of the wire and the range of the load which can be measured can be changed freely. This load dynamometer uses uninsulated electric-resistance wire coiled around the stressed cylinder without any adhesive cements. Therefore, the rate of heat dissipation from the wire is greater than that in the ordinary bonded strain-gage installation which is insulated by adhesive cements. This permits the gage current I_g to be larger for this type of load dynamometer than for a strain meter insulated by adhesive cements.

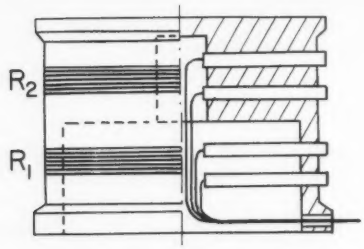

Fig. 4—One type of stressed body

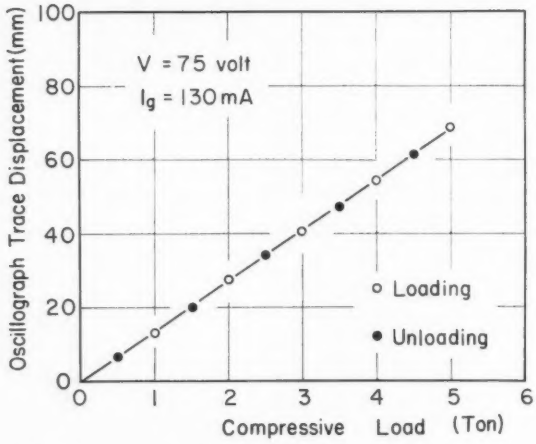

Fig. 5—Calibration result of the load dynamometer

As a result, this load dynamometer is very stable and the measuring can be done without any amplifier, because of the large bridge unbalance of this measuring unit.

Electric Circuit Used, Cylindrical Stressed Body and Calibration

The electric resistance of wire changes with changes in its temperature caused by changes in gage current. Therefore, resistance wire[9] which has a small temperature coefficient of electric resistance (such as Advance wire, Constantan, manganese wire, etc.) is preferable and, in general, a temperature-compensation circuit is made. Figure 2 shows one example of such an electric circuit. It is desirable from the viewpoint of temperature compensation that the strained (active) resistance R_1 being loaded and the unstrained (dummy) resistance

R_2 for temperature compensation are located as close as possible to each other.

The Wheatstone bridge in Fig. 2 is usually used in the condition that all resistance wires, R_1, R_2, R_3 and R_4, have the same value R of electric resistance. The following factors in the designing of electric circuit must be considered in order to increase the bridge unbalance current I_0:

1. It is desirable that the electric resistance R of the resistance wire be larger than the internal resistance R_i of the measuring instrument.

2. Current in the gage I_g is to be large. It is necessary for this purpose that the diameter of the electric resistance wire be large. This is made possible by constructing the stressed body in cylindrical shape and winding the electric resistance tightly around its circumference.

3. The strain value ϵ of the resistance wire caused by external load must be large. It is necessary for this purpose that a material which is capable of large elastic strains be used as a cylindrical stressed body.

Some examples of such stressed bodies are shown in Figs. 3 and 4. The arrangement for a compression test of a cylindrical specimen using this dynamometer is shown in Fig. 3. In Fig. 3, R_2 is coiled around a collar which is not strained during loading. On the other hand, the stressed body, shown in Fig. 4, has two cylindrical parts having different cross-sectional areas. R_1 must be coiled around the small cross-sectional part A_1 of the cylinder, and R_2 must be coiled around the large cross-sectional part A_2. Because R_2 must not be strained appreciably during loading, the ratio of A_2 to A_1 must be very large. In particular, the cylindrical stressed body of Fig. 3 has better character than that of Fig. 4, because it is certain that R_2 of Fig. 4 is strained during loading more or less dependent upon the ratio of A_2 to A_1. However, the cylindrical stressed body of Fig. 4 has several merits such as ease in design, machining and construction, etc.

In general, the resistance wire is tightly coiled with small pretension into a very shallow spiral groove which is machined on the surface of stressed body, and both ends of the resistance wire are fastened by soldering or brazing to the small screws installed in the stressed body, as shown in Figs. 3 and 4. The following procedures are usually applied in order to prevent electrical shorting between stressed body and resistance wire:

1. Surface treatment of stressed body, for example, Parkerizing coating for carbon steel, aluminum oxide coating for aluminum and its alloys.

2. Spraying with an insulation enamel of synthetic resins such as Bakelite, silicone and polyester varnishes, and for better stability of the insulation effect, curing at elevated temperature.

The detailed information of the stressed body, which is designed for the measuring of compressive loads up to 5 tons, and the related measuring conditions are as follows:

1. Stressed Body.
   (*a*) Material: carbon steel (C = 0.5–0.6%).
   (*b*) Heat treatment: tempering after quenching.
   (*c*) Surface treatment: Parkerizing coating and then spraying on Bakelite varnish and further curing at 180° C for 1 hr.
   (*d*) Dimensions: original height H_0 = 60 mm; original diameter D_0 = 46 mm.
2. Electric Circuits.
   (*a*) Resistance R_1 and R_2 = 200 ohms each. (Advance wire having diameter of 8×10^{-2} mm).
   (*b*) Resistance R_3 and R_4 = 200 ohms each. (Manganese wire which is coiled around Bakelite bobbin without any electric induction).
   (*c*) Gage current I_g = 130 ma.

The above data were stated as one example of this new load dynamometer and measuring conditions used.

In regard to the calibration of this load dynamometer, the bridge unbalance currents for applied compressive load are accurately measured for each respective gage current I_g by a precision ammeter (μa). To use this dynamometer for measuring the yield point of high-polymer solids, however, the calibration was performed with the output recorded by means of electromagnetic oscillograph with a *D* type vibrator installed. The relationship between the compressive load applied on the stressed body and the displacement of spot image on the rotating drum of the oscillograph was calibrated for the case of V-75v dc input, I_g = 130-ma. The calibration data for both loading and removal of load are plotted in Fig. 5. The calibration results are satisfactorily linear, and no hysteresis loop or zero shift was observed. All elements of this load dynamometer have been kept indoors without special environment and recalibrated in the same manner twice, at six months, and again at one year after the initial calibration. The linearity and reproducibility of the later calibration results were also satisfactory. In spite of severe environmental conditions, in particular, the considerable humidity change in Tokyo through the year, the calibration remained stable. This stability contributes to the reliability of the measurements made. The small pretension of resistance wire, the surface treatment of stressed body and the insulating enamel are important factors in determining the accuracy, stability, reliability, zero shift, etc., of this load dynamometer.

The bridge unbalance current, which is induced by the applied load and which agitates the rotating action of *D* type vibrator in the oscillograph, is larger than that in any other known type of strain gage utilizing resistance wire. Thus, this new load dynamometer, utilizing electric-resistance wire, is simple, successful and able to measure various dynamic or transient changes in load under ordinary environments without inertia effects or amplification.

Application to Measurement of the Yield Points of High-polymer Solids

At room temperature, the yield phenomenon of

high-polymer solids appears tardier than that of mild steel, yet with evident upper and lower yield points.[10] Because this yielding of high-polymer solids is a new interesting phenomenon[11] and because the true stress-strain curve must be correctly determined for the portion of the curve beyond the upper yield point where the load is decreasing, this load dynamometer is applied to the measuring of the yielding behavior of high-polymer solids in various rates of strain.

As shown in Fig. 3, a cylindrical test specimen (d/h = 20 mm/20 mm) is arranged in series with the stressed body. The strain in the stressed body is very much less than that of the test specimen and the stressed body can immediately respond to an applied load without inertia effects. Therefore, the test specimen is compressed at a constant rate of strain if the upper and lower plates in Fig. 4 move at a constant rate. The rate of stroke of the upper and lower plates is easily controlled by the change of the resistance in d-c motor circuit. Thus the rate of strain, which is kept constant during any one test, is varied for different tests.

The change of compressive load is automatically recorded on the drum of the electromagnetic oscillograph which has D type vibrator and is connected electrically to the stressed body through the Wheatstone bridge. The velocity of rotation of the drum should correspond with the respective rates of strain in the test specimen.

Two thermosetting cast resins (epoxy and diallylphthalate homopolymer) are used as test specimens. Both cast resins have upper and lower yield points at room temperature, although yield points of epoxy resin are more pronounced[12, 13] than those of diallylphthalate homopolymer (DAP) due to the difference of internal molecular structure. The initial diameter, d, of the cylindrical test specimen is 20 mm so that the compressive load at the upper yield point can be adequately recorded by the 5 ton stressed body, as stated in the previous section. The initial height, h, of the specimen is also 20 mm and, thus, the ratio of initial height to initial diameter of the specimen is always one for all compression tests at each different rate of strain. This is necessary because the initial configuration of the test specimen influences the stress-strain curves in compression. The oscillograms obtained from the compression tests of both cast resins at various rates of strain are shown in Figs. 6 and 7. In Figs. 6 and 7, r (%/sec) is the rate of compressive strain in the test specimen, n_D(rpm) is the rotating velocity of the drum in oscillograph and v_D (mm/sec) is the peripheral velocity of the drum in oscillograph. These results of epoxy resin and DAP are plotted in Figs. 8 and 9, respectively, taking nominal compressive stress (kg/mm^2) as ordinate and nominal compressive strain (%) as abscissa. The upper yield point of both resins is influenced by the rate of strain.[14]

The influence of the initial configuration of cylindrical test specimens on the stress-strain curves in compression has been explored in the same way. Test specimens having the same initial diameter, d = 20 mm, but with various initial heights, h, were machined. These specimens, having various

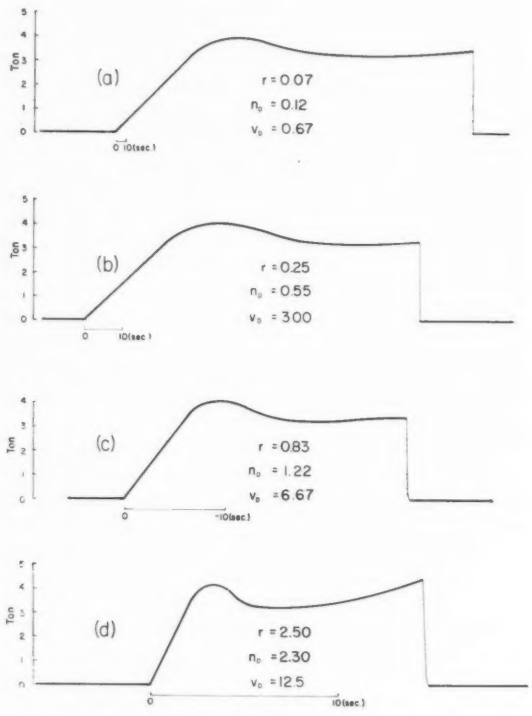

Fig. 6—Oscillograms in the compression test of epoxy cast resins in various rates of strain

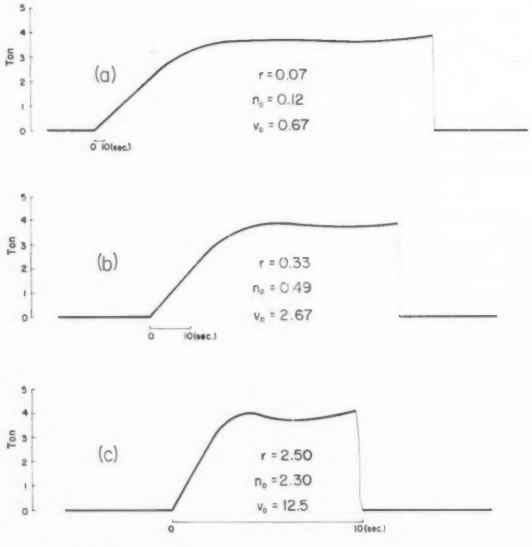

Fig. 7—Oscillograms in the compression test of DAP cast resins in various rates of strain

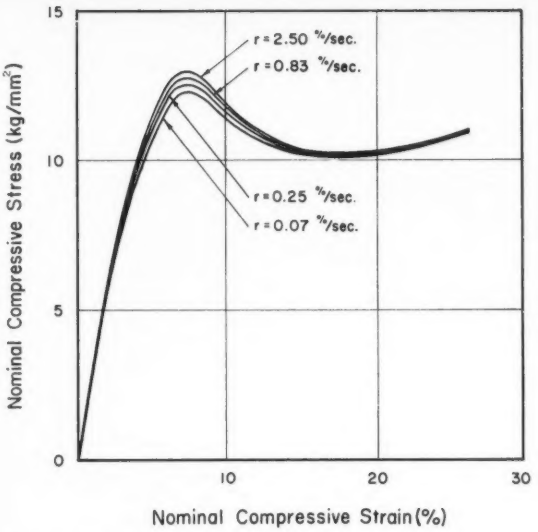

Fig. 8—Compressive stress-strain curves of epoxy cast resins in various rates of strain

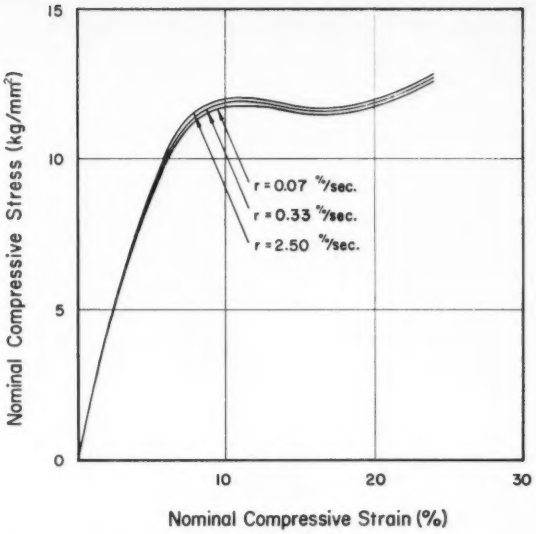

Fig. 9—Compressive stress-strain curves of DAP cast resins in various rates of strain

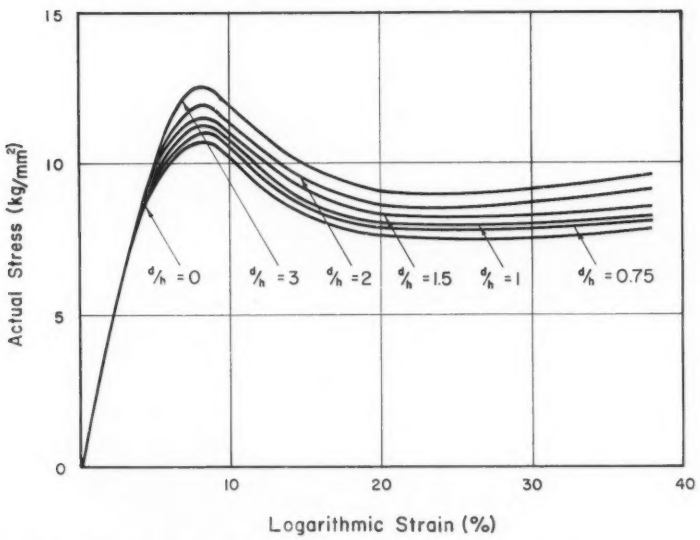

Fig. 10—Compressive "Fliess-kurve" of epoxy cast resins having various initial configuration (d/h)

values of d/h were compressed at the constant rate of strain, $r = 0.07\%$/sec, controlled by the change of the resistance of d-c motor circuit. In these compression tests, the oscillograms have been recorded in the same manner as previously shown in Fig. 6 and in Fig. 7. In this case, celluloid, which has also upper and lower yield points at room temperature, is used as a representative of linear polymer solids. The results obtained from these oscillograms for the tests of epoxy cast resin and celluloid are plotted in "Fliess-kurve" form (actual stress-logarithmic strain curves), as shown in Figs. 10 and 11, respectively. The "Fliess-kurve" in the case of $d/h = 0$, has been evaluated by an extrapolation method. It is easily understood that the frictional resistance of both compression ends, in particular, in the case of disc-type specimens affects the stress-strain curve observed. In particular, it affects considerably their upper yield points. These results have the same tendency as

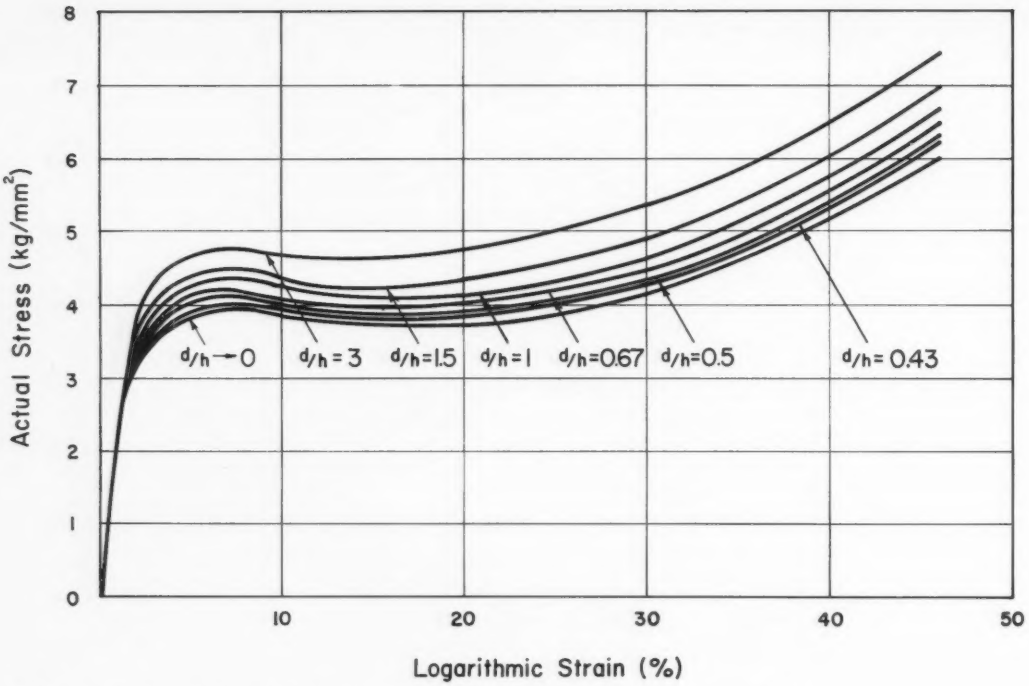

Fig. 11—Compressive "Fliess-kurve" of celluloids having various initial configuration (d/h)

G. Sachs' test[15] on the compression of metal specimens.

The upper and lower yield points of mild steel disappear at high temperatures. Also the upper and lower yield points of high-polymer solids, in particular, of the cast resin of three-dimensional cross-linked polymers disappear[10, 13] completely in the temperature range of entropy elasticity, at which secondary bonding forces are dissociated. Consequently, the secondary bonding forces which are greatly related with the mechanical properties of high-polymer solids at room temperature, must be considered as an important factor of the occurrence of the yield phenomenon. The experimental fact that the yielding of epoxy resin is more evident than that of DAP, as shown in Figs. 8 and 9, is clearly understood by the fact that the secondary bonding forces of epoxy cast resin, which appear to be mainly hydrogen bond, are simpler and more homogeneous than those of DAP. On the basis of the inertialess measuring of the yield points of high-polymer solids by this new load dynamometer, it has been proved that the tardy yield point is the peculiar characteristic of high-polymer solids at room temperature. This tardy yielding may be expected from the complicated molecular structure of high-polymer solids by heteropolymerization, while the jumping yield phenomenon of mild steel at room temperature is caused by the simple honeycomb structure consisting of soft ferrite and hard cementite.

Acknowledgment

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Some Aspects of Bending Brittle Materials

Information of value to design engineers faced with problem of recommending materials for applications involving bending is presented in this technical note

by Conan H. Spaderna

In spite of advanced stress analysis, it is sometimes not easy to recommend a certain material for a design where bending is involved. To try all materials under operational conditions would be prohibitive if the best material should be picked out of a large group. Usually, the designer has to rely upon a few test data, though he knows well that these data are correct only for special test conditions. He needs corrections, however, which are not available. This is even worse with those brittle materials which have a smaller modulus of elasticity in tension than in compression, a different Poisson's ratio in tension and in compression and, often, no linear relationship between stress and strain.

W. H. Duckworth has shown in a practical way how to deal with the first difficulty.[1] He uses two-point loading on rectangular beams. The space between the two load edges is wide enough to accommodate a strain gage which measures the compressive strain ϵ_c. On the opposite side of the beam, another strain gage measures the tensile strain ϵ_t. With F = modulus of rupture, computed in the conventional way, the *true* bending strength S becomes:

$$S = F \cdot \frac{\epsilon_t + \epsilon_c}{2\epsilon_t} \qquad (1)$$

The correction of F, thus effected, is generally between -1 and -12%.

However, over different support spans, the true bending strength S varies. Two materials, which gave the same results over the testing span, may easily differ by 20% in strength at a shorter span, called for by the design. How to compute the bending strength for any practical span will be shown later.

It has to be stated that eq 1 applies only to materials with a *linear* stress-strain relationship. For the many other materials, such as cast iron, another procedure is proposed.

A New Way of Computing the True Bending Strength

Three strain gages are required. One, lengthwise on the tensile side, one, lengthwise on the compressive side of the beam, and one, perpendicular to the latter. Thus, ϵ_t, ϵ_c and Poisson's ratio in compression, μ_c, can be measured.

Conan H. Spaderna was associated with Norton Co. at the time that this technical note was prepared.

The true bending strength S is, if F = modulus of rupture,

$$S = F\left(1 - \frac{\mu_c^2 \epsilon_c^3}{\epsilon_t^3}\right) \qquad (2)$$

here:

ϵ_c = unit strain in compression.
ϵ_t = unit strain in tension.

The two strains are taken at the instant before the crack springs up, which is easily found on the record.

This method is independent of linear load-strain relationships and works also with round rods. It was used on materials, having almost circular load-strain curves. With rectangular beams of straight load vs. strain linearity, the results of eq 1 and 2 usually differ by less than the reading error of the test. For extremely short spans, eq 1 makes $S > F$ and thus cannot be used.

Deduction of Eq 2

This method starts with the assumption, that the shift of the neutral axis can be computed from external observations on the beam. The neutral axis in bending is defined here as the fiber of zero normal strain. With several strain gages equally distributed over the depth of the beam, it is possible to locate the unstrained fiber. The measured strains, plotted over the height from the bottom, yield lines which intersect at the height of the neutral axis.

Empirically, the writer found that the neutral axis is lowered by

$$\frac{Kh}{2} = \frac{\mu_c^2 \epsilon_c^3 h}{2\epsilon_t^3} \qquad (3)$$

Here, h is the depth of the beam (see Fig. 1). Conventionally, the modulus of rupture, F, is computed on the assumption that the neutral axis stays in the middle of the cross section, at the distance $h/2$ from the bottom edge. $F = M/I \cdot h/2$, if

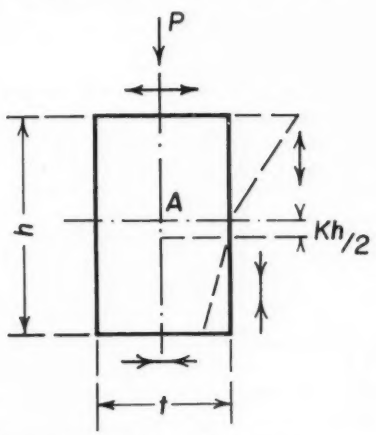

Fig. 1—Cross section of a beam in bending

M = maximum bending moment, and I = moment of inertia.

Introducing K from eq 3, we obtain eq 2. Specifically, for rectangular beams:

$$S = \frac{3Py(1 - K)}{th^2} \qquad (4)$$

For circular rods:

$$S = \frac{8 \cdot Py(1 - K)}{\pi d^3} = \frac{2.546Py}{d^3}(1 - K) \qquad (5)$$

In eqs 4 and 5 are, with all dimensions in inches:

P = total load in pounds.
y = moment arm between load edge and support.
t = thickness of the beam, width.
h = height of the beam, depth.
d = diameter of the rod.
$K = \mu_c^2 \cdot \epsilon_c^2 / \epsilon_t^3$, see eq 2.

Timoshenko[2] stated $S = F(1 + 0.0289\, h/y)$ with h = depth of the beam, y = moment arm. Neither Duckworth's nor the writer's results agree with this simple formula which is not based on the relationship of top and bottom strains.

The Influence of the Setup

Figures 2 and 3 show bending setups, known as "two-point" and "one-point" loading. In either case, the load is resisted by a reaction from the support edge. The two forces enter at an angle β (stiffness angle) to the support span, which angle is defined by: $\tan \beta = h/y$. Here, h is the depth of the beam, y is the moment arm.

Measurements by the writer on cast iron, refractories, concrete, etc, lead to this relationship:

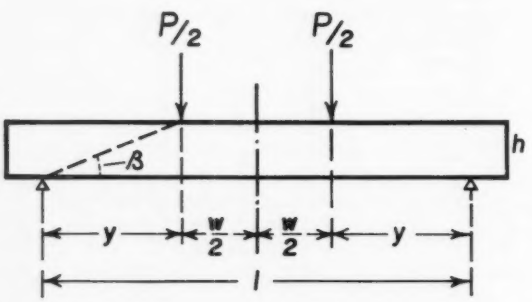

Fig. 2—Beam with two-point loading

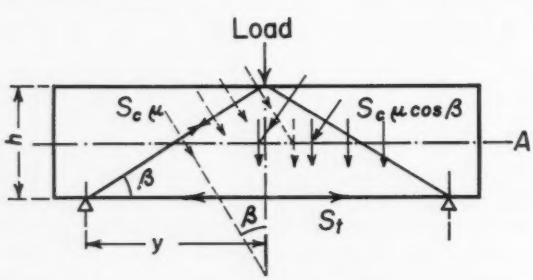

Fig. 3—Stresses in a beam in bending under midspan load

$$S = S_t + S_c\, \mu_t \cos \beta \qquad (6)$$

S = true bending strength, psi.
S_t = tensile strength from tensile tests, psi.
S_c = resistance to crushing, psi.
μ_t = Poisson's ratio in tension.
β = stiffness angle.

Eq 6 holds within these limits of $\tan \beta = h/y$:

$\tan \beta = 0.2$—1.0 for two-point load
$\tan \beta = 0.1$—0.2 for one-point load

If all components are measured at the same temperature and loading rate, S can be predicted for design. Generally, with rising β, the bending strength S becomes smaller, depending upon $S_c\mu_t$. Therefore, no general law for all brittle materials can state S as a function of the modulus of rupture, F, and of the support span.

Examples

1. Concrete slabs 2.6 × 6 × 15 cm were bent flat on a 14 cm span by two point loading. Load span: 2.5 cm. Loading rate 7 psi/sec, $h/y = 0.45$, room temperature mean results: $F = 788$ psi, $S_t = 227$ psi, $S_c = 3416$ psi, $\mu_t = 0.165$. From eq 6: $S = 227 + 3416 \times 0.165 \times 0.91 = 739$ *psi*. From eq 2: $S = 788.(1 - 0.31^2 \times 0.76^3) = 788.(1 - 0.096 \times 0.442) = 741$ *psi*. From eq 1: $S = 788 \times 0.93 = 743$ *psi*.

2. Cast-iron rods, OD = 3 cm, support span 60 cm, gave under midspan load: $F = 67{,}500$ psi; $h/y = 0.1$ is within limits. $S_c = 125{,}000$ psi, $S_t = 35{,}200$ psi, $\mu_t = 0.25$. From eq 6: $S = 352{,}000 + 125{,}000 \times 0.25 \times 0.999 = 66{,}400$ *psi*.

Physical Meaning of Eq. 6

Assuming equilibrium of forces at the instant just before breakage, Fig. 3, the atom lattice offers a resistance, which offsets Poisson's forces $S_c\mu_t$, shown by broken arrows. Projection of this resistance into a cross section is brought about by the factor $\cos \beta$. This is the strength of the cross section against the tangential stress, attacking at the same time as the normal tensile stress S_t. Thus, the tensile strength and the tangential resistance $S_c\mu_t \cos \beta$ add up to the total bending strength S. The resistance forces $S_c\mu_t$ overlap partially with one-point load, raising the bending strength over what it would be with two-point load.

To explain Poisson's lattice forces, $S_c\mu_t$, it is theorized that atom groups getting closer, team up their repulsive forces at right angles to the direction, in which they are pushed together and a tension results across the direction of push. If the $S_c\mu_t$ forces cross, Fig. 3, complications arise. To forgo these complications, the limitations of $\tan \beta$ were made. By the same token, it is desirable to have a large ratio of loading span over depth, $w/h \geqq 1$, especially, when testing with a steep angle β.

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Photodot Investigation of Plastic-strain Pattern in Flat Sheet with a Hole

Dot-pattern method used for measuring strains in a thin flat 24ST-3 aluminum sheet with a central hole, unidirectionally loaded into plastic region of material

by P. S. Merrill

ABSTRACT—The strain pattern was measured, using the photodot method, in a thin flat 24ST-3 aluminum sheet with a central hole, unidirectionally loaded into the plastic region of the material. Test results are presented and the accuracy of the photodot method is discussed.

Introduction

The strain pattern in a thin flat sheet with a central hole, unidirectionally loaded, as in Fig. 1, beyond the elastic limit of the material, has been experimentally investigated at the point of maximum strain and along the transverse axis by several investigators.[1-3] It was shown that, as the load increased, the strain-concentration factor at the edge of the hole increased to a very large value while the stress-concentration factor decreased to a value near unity.

The large strains at the hole edge cause some question about the distribution of strains elsewhere in the sheet. The strains along a longitudinal line through the hole edge must decrease to very low values to allow the over-all longitudinal displacement to be the same as the over-all displacement at the sheet edge. Also, as pointed out by Nadai,[4] the plastic region may extend along lines at an angle to the transverse axis rather than along the transverse axis.

Objectives

The object of this investigation was to determine experimentally the longitudinal strain pattern, in the unidirectionally loaded thin flat sheet with a central hole, for strains beyond the elastic limit of the material, without elaborate special equipment. The photodot method used for measuring strains is evaluated for accuracy, and methods of improving the accuracy are discussed.

Description of Test Piece

The test piece, Fig. 1, was geometrically similar to the 24ST-3 aluminum test piece used by Griffith,[2] except for thickness.

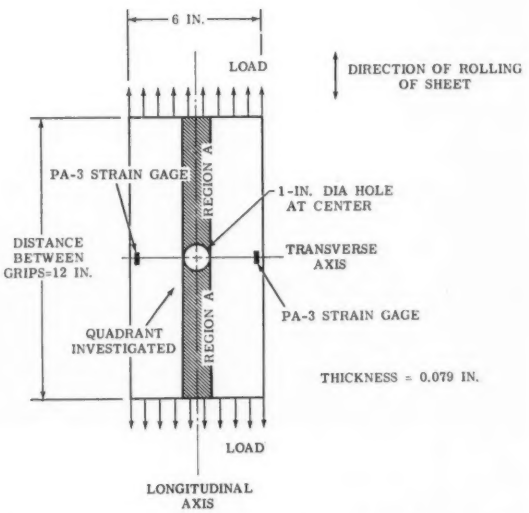

Fig. 1—Test specimen (24ST-3 aluminum sheet with hole)

The tensile properties of the material, Fig. 2, were obtained from tensile-test strips cut from the sheet adjacent to the test piece. For the strip cut parallel to the direction of rolling, the stress-strain relationship curve contained a sharp knee; but for the strips cut at 45 and 90 deg to the direction of rolling, the knees were less pronounced. Intermittent localized slip was evidenced by flow markings on the surface of the tensile strips after testing.

Test Equipment

General

A dot pattern on the test specimen was photographed before and during loading. Changes in distances between dot edges were computed from a comparison of negative measurements and converted

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to strains. Dots were selected instead of the usual grid lines for the following reasons:

1. To eliminate the necessity of measuring to possibly indistinct grid line intersections.
2. To allow easy measurement on the diagonal for the determination of principal strains.
3. To reduce the manufacturing complication.

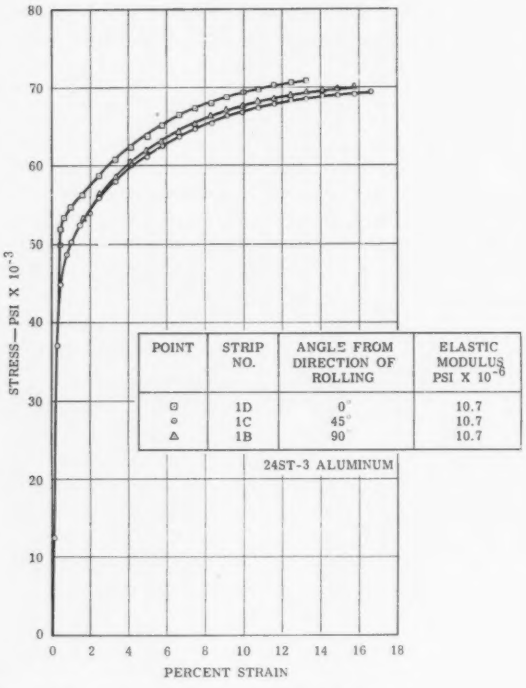
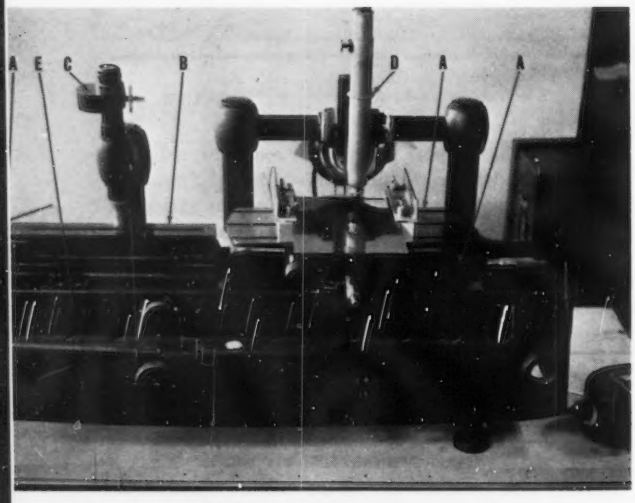

STRESS—PSI X 10^{-3}
PERCENT STRAIN
POINT | STRIP NO. | ANGLE FROM DIRECTION OF ROLLING | ELASTIC MODULUS PSI X 10^{-6}
□ | 1D | 0° | 10.7
○ | 1C | 45° | 10.7
△ | 1B | 90° | 10.7
24ST-3 ALUMINUM

Fig. 2—Stress-strain characteristics. (Actual measurements)



A. Moving table, **B.** Scale, **C.** Scale-measuring microscope, **D.** Table-positioning microscope, **E.** Frame.

Fig. 3—Comparator

Dot Pattern

The dots were approximately 0.005 in. in diameter and were arranged in rows and columns 0.050 (±0.0015) in. apart. The 6- by 6-in. dot pattern was approximately centered on the hole center.

A master negative was photographically produced from an engraving of the dot pattern. The dot pattern was applied to the test specimen essentially as described by Miller[1] and others[5–7] using the standard techniques of the first part of a process for manufacturing Westinghouse nameplates. A light-sensitive emulsion was poured on the specimen; excess emulsion was removed by spinning on a rotating table. After the emulsion was dry, the master negative of the dot pattern was placed over the emulsion and the two were exposed to an arc light. The unexposed soluble emulsion was removed in water containing aniline dye, leaving the insoluble dots dyed blue. The dot pattern was placed on several test specimens and on two tensile strips.

Spinning the test specimens caused two parallel streaks in the emulsion leading tangentially from the hole edges. The quadrant selected for measurements did not contain these streaks.

Comparator

The comparator for measuring changes in the distances between dot edges is shown in Fig. 3. The locations of points positioned under the positioning microscope were read on the scale-reading microscope with a least count of 0.001 mm.

In almost all cases, strains were obtained from successive readings to dot edges on the operator's right in a line parallel to the direction of strain measurement. Errors in a single dot position reading appeared as a pair of strain errors, one error positive and the other negative.

Photographic Equipment

Figure 4 shows the camera and lighting arrangement used for photographing the dot pattern. Figure 5 is an enlargement of one of the dot pattern negatives.

The camera lens was a Leitz Summar, focal length 120 mm, F4.5, uncoated, manufactured prior to 1930. A Kodak No. 15 blue Wratten filter was used in front of the lens to cause the purple dots on the test piece to appear darker to the negative.

The film was Kodak Kodalith Orthoplate, 5 by 7 in. This was an extremely high contrast, high resolution glass plate that was sensitive to the green and blue parts of the light spectrum. The high contrast characteristics of the plate caused overexposed dots to be reduced in size on the negative rather than to appear gray. Glass plates were selected for their known dimensional stability compared to the uncertain stability characteristics of films. For better negative-to-negative uniformity, the plates for a particular test were obtained from the same box and were all developed simultaneously.

The camera was focused to give approximately

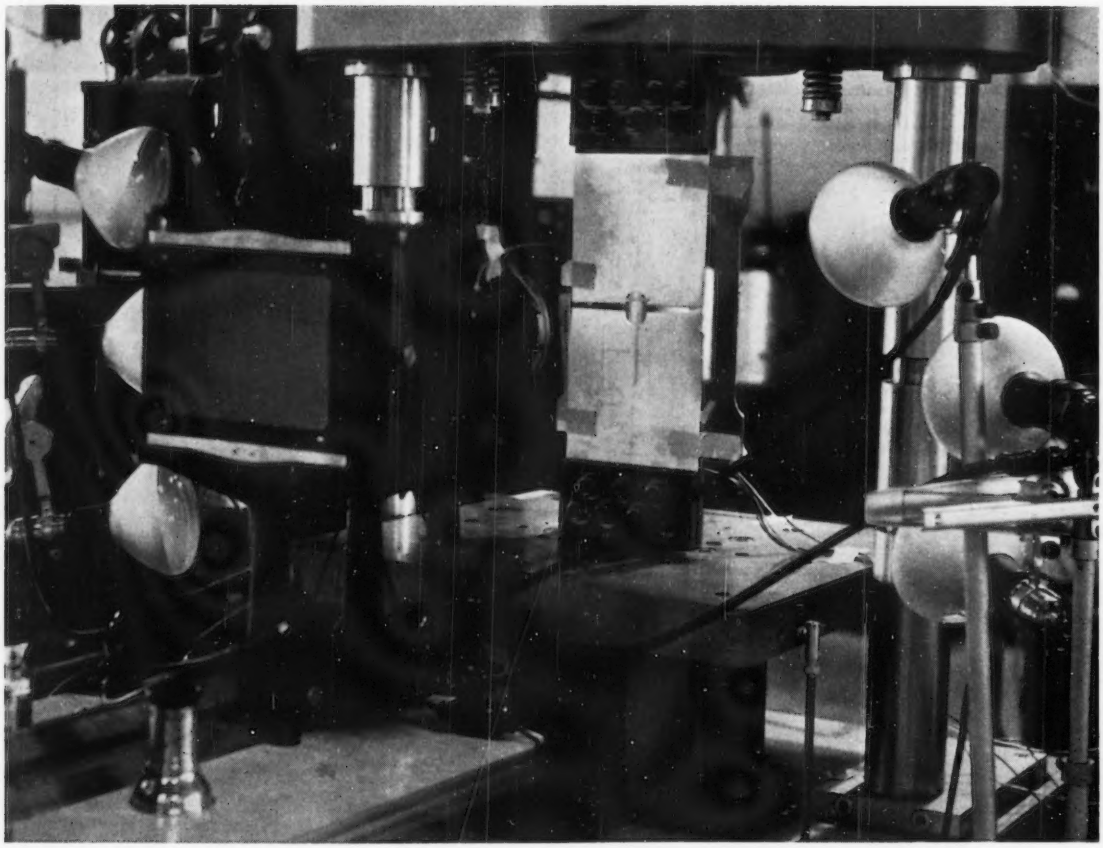


Fig. 4—Tensile-test equipment

200% magnification. To check the exact magnification, 0.005-in. thick aluminum reference strips containing a row of dots were glued at one end to the surface of the test specimen.

For negative measurements, the negative plates were raised above the comparator table and a white card was tilted to diffuse light through the transparent dots. The criterion for microscope hairline alignment on the dot edges was the appearance of a slight glow on the side of the hairline away from the dot. The strains obtained from the measurements of the negatives were corrected for variations in magnification, using distances between dots on the aluminum reference strips.

Strain Gages

Two Type PA-3 strain gages were placed on the transverse axis of the test specimen, 0.5 in. from the sheet edges to check uniformity of loading.

Accuracy Estimates

The factors known to contribute to the over-all errors of the photographic strain-measuring system are listed in Table 1. To evaluate the cumulative uncertainty intervals[8] of the strains measured with

TABLE 1—ESTIMATED SOURCES OF ERRORS IN STRAIN MEASUREMENT USING PHOTOGRAPHIC SYSTEM

Approximate Magnification on Negative = 200%; Gage Length on Specimen = 0.050 In.

Source of Error	Estimated Uncertainty Intervals, 95% Confidence: Cumulative Strain, %	Individual Strain, %	Individual Measurement on Specimen
Tensile Strip			
1. Clearness of dot edges			
2. Errors inherent in the comparator			
3. Operator errors	0.15	0.15	0.000075
4. Sharpness of reproduction of dot on negative			
5. Nonuniformity of negative emulsion and development (causing dots to change size)	0.25	0.20	0.000100
6. Lens distortion			
7. Nonuniformity of light transmittance to negative causing changes in dot sizes	0.30	0.17	0.000085
Sheet with Hole			
Total uncertainty interval (except where noted)	0.4		0.000200

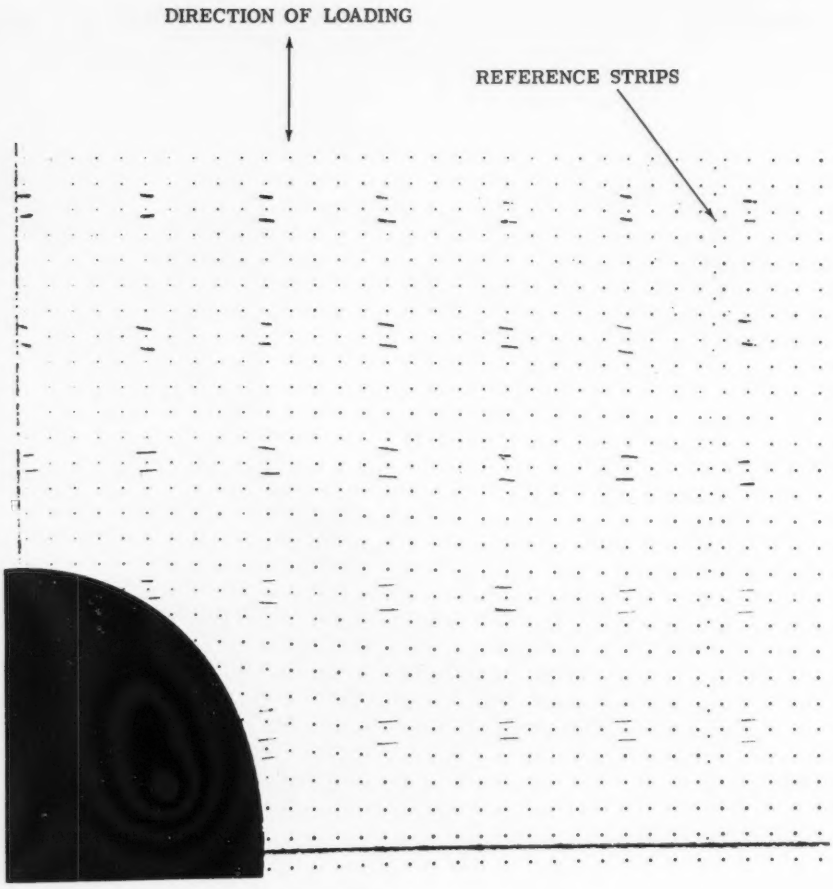


Fig. 5—Enlarged view of a portion of dot-pattern negative taken at 24,400 lb load

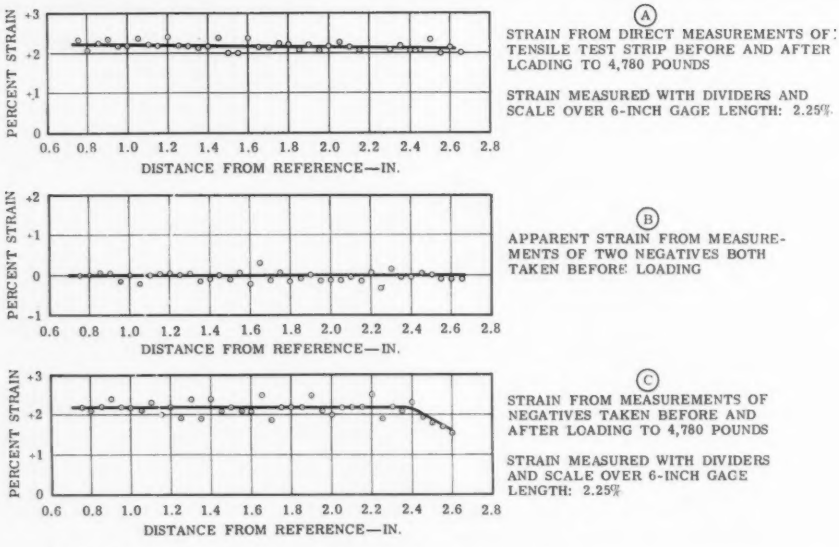

PERCENT STRAIN
+3
+2
+1
0
0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4 2.6 2.8
DISTANCE FROM REFERENCE—IN.
Ⓐ
STRAIN FROM DIRECT MEASUREMENTS OF TENSILE TEST STRIP BEFORE AND AFTER LOADING TO 4,780 POUNDS
STRAIN MEASURED WITH DIVIDERS AND SCALE OVER 6-INCH GAGE LENGTH: 2.25%
PERCENT STRAIN
+2
+1
0
-1
0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4 2.6 2.8
DISTANCE FROM REFERENCE—IN.
Ⓑ
APPARENT STRAIN FROM MEASUREMENTS OF TWO NEGATIVES BOTH TAKEN BEFORE LOADING
PERCENT STRAIN
+3
+2
+1
0
0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4 2.6 2.8
DISTANCE FROM REFERENCE—IN.
Ⓒ
STRAIN FROM MEASUREMENTS OF NEGATIVES TAKEN BEFORE AND AFTER LOADING TO 4,780 POUNDS
STRAIN MEASURED WITH DIVIDERS AND SCALE OVER 6-INCH GAGE LENGTH: 2.25%

Fig. 6—Strains of tensile strip obtained by direct measurement and by measurement of photographs of dot pattern

the system, strains from dot patterns on tensile strips were measured directly and with photographs before and after loading into the plastic region of the material. From the results shown in Fig. 6, the cumulative uncertainty intervals listed in Table 1 were estimated. The sums of the squares of the individual uncertainty intervals totaled the squares of the cumulative uncertainty intervals.

The uncertainty intervals from the direct measurement of the tensile strips, Fig. 6A, contained only factors 1, 2 and 3. The actual uncertainty interval was 0.3% strain with 95% confidence, but this was equivalent to a 0.15% uncertainty interval on the double-sized negatives.

The cumulative uncertainty interval for factors 1 through 5 were estimated from strain measurements from two negatives taken at zero load, Fig. 6B. Uncertainty intervals due to factors 6 and 7 appeared after the specimen was elongated and the dots moved past the camera to areas of different lens distortion and light transmittance.

The over-all uncertainty interval of the tensile-strip measurements was estimated from strain measurements of negatives of the tensile strip taken before and after loading into the plastic region of the material. From the results shown in Fig. 6C, it was estimated that the uncertainty interval was 0.3% strain with 95% confidence. Readings of points less than 0.4 in. from the negative edges were considered unreliable.

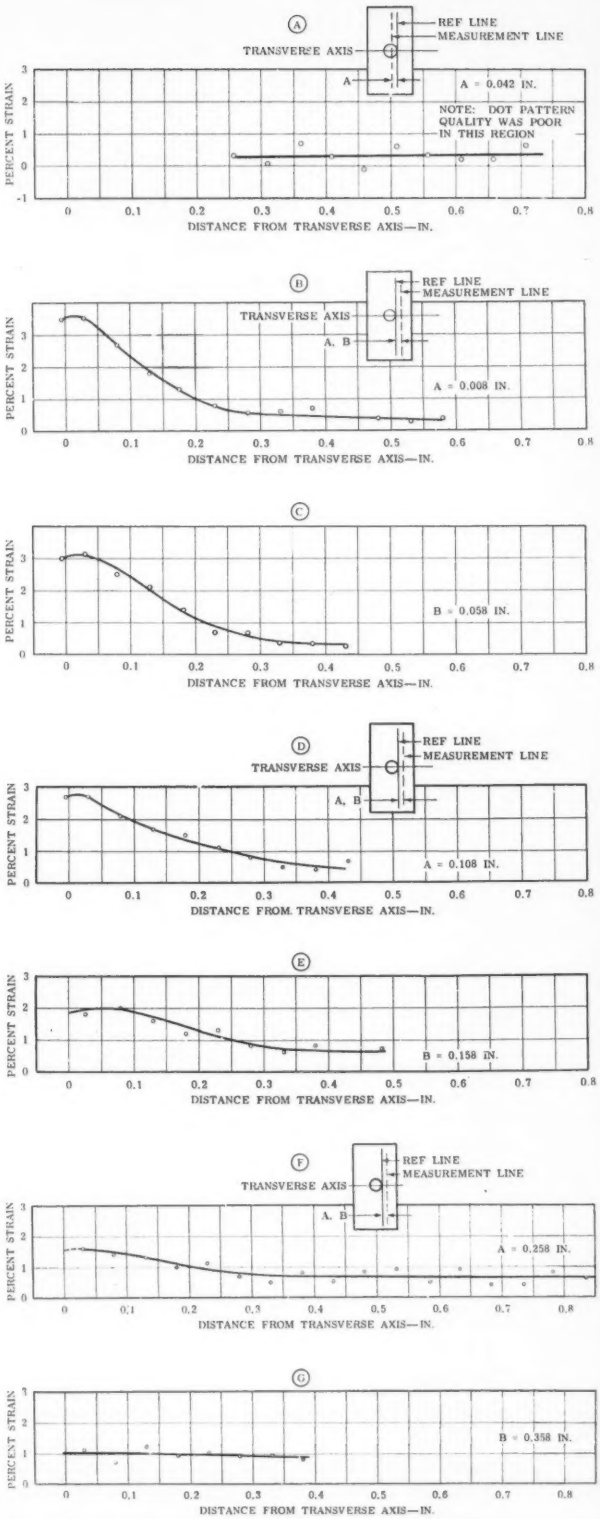
On the actual test specimen, the dot-pattern quality was poorer than on the tensile strip. Therefore, the uncertainty interval of the strains on the test specimen was estimated to be generally 0.4% strain with 95% confidence.

The superior sharpness of a dot observed on the master negative showed that the dot quality observed on the specimen and measuring negative were poor compared to the capabilities of the Kodalith Ortho-plate.

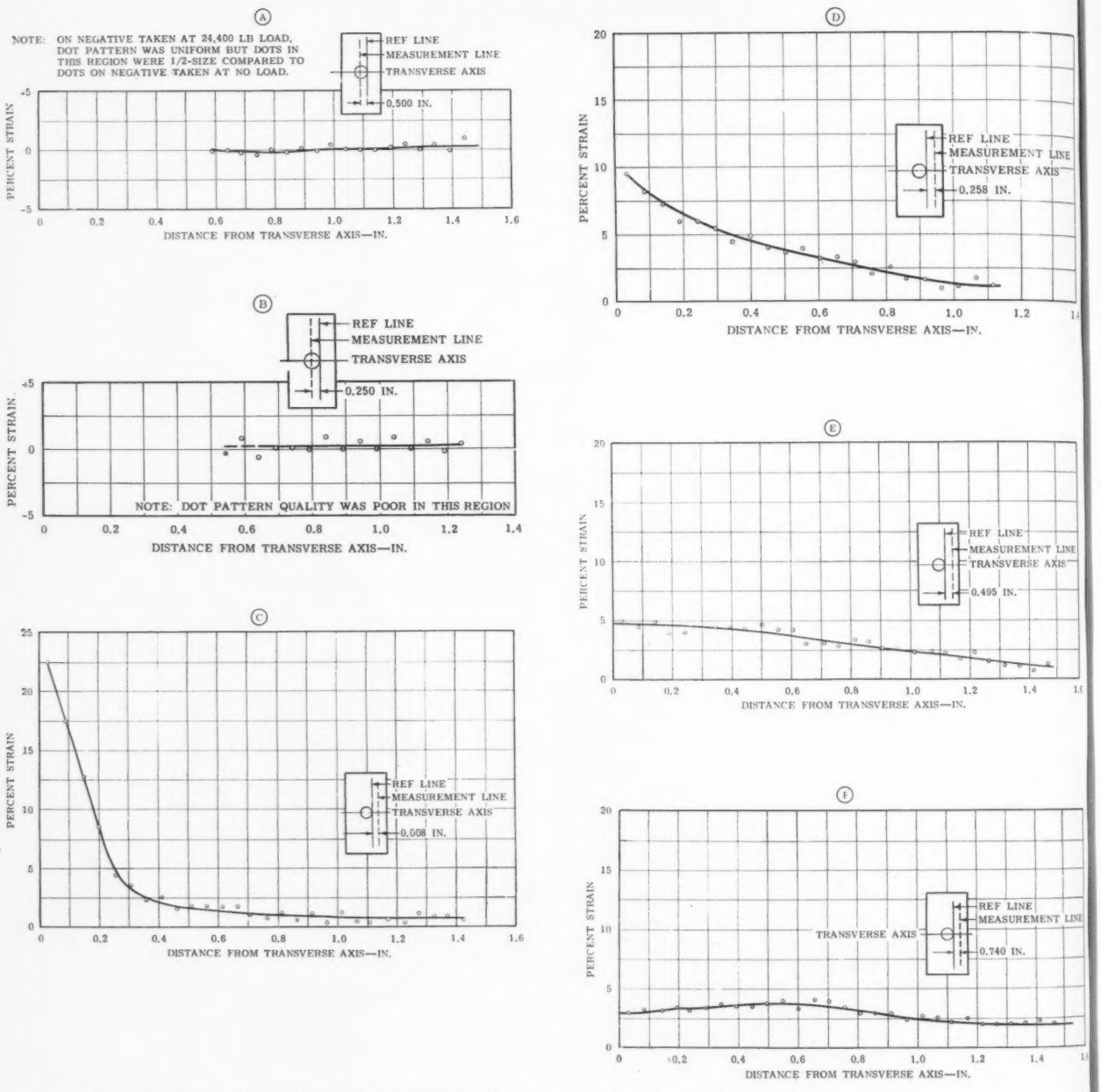
Uneven exposure of the dots was evidenced by nonuniform dot sizes on the negatives (see Fig. 5.) Much of the uneven exposure, especially at the edges of the negatives, appeared to be independent of the positions of the dot pattern on the negatives. Therefore, it is believed that much of the uneven exposure was caused by nonuniform light transmittance to the negatives attributable to lens design, absence of low-reflectance lens coatings, and reflections inside the camera. The remaining causes of uneven negative exposure were believed to be nonuniformities in the lighting and color of the dots, and reflections from the shiny surface of the test piece.

Procedure and Results

Strains were measured along longitudinal lines for loads of 12,850; 16,800; 20,800 and 24,400 lbs. Typical results for loads of 20,800 and 24,400 lbs. are presented in Fig. 7 and 8. Maps of the strains are presented in Fig. 9. The 24,400-lb. point was not



Figs. 7A—7G—Longitudinal strains along longitudinal lines. (Plate with hole, loaded to 20,800 lb)



Figs. 8A–8F—Longitudinal strains along longitudinal lines. (Plate with hole, loaded to 24,400 lb)

mapped because too few points were obtained.

Although the relative gage lengths of Griffith's similar 24ST-3 specimen were 25% larger than the gage lengths of this investigation, agreement between the two results was obtained as predicted from the laws of similitude.[9]

Except for region A (Fig. 1) and for the condition shown in Fig. 8F, the maximum longitudinal strains along longitudinal lines were located very near the transverse axis. In Fig. 8F, for a load of 24,400

lbs., the strain curve on a longitudinal line 0.740 in. from the hole edge is nearly flat with a peak centered approximately 0.6 in. from the transverse axis. No longitudinal plastic strains were encountered in Region A.

The average difference between the strain readings from the two PA-3 strain gages was 3.1%; the maximum difference was 7.5%.

The failure closely followed the transverse axis. The plane of the failure was at an angle of approximately 45 deg from the two faces of the test specimen.

Discussion

Strain Patterns

From Fig. 9, it is seen that for loads up to 20,800 lbs. the region of strains larger than 1.5% was confined to a region within 0.225 in. of the transverse axis and 0.275 in. of the hole edge. However, as shown in Fig. 8, for loads between 20,800 lbs. and failure at 25,700 lbs., the region of strains larger than 1.5% extended rather far into the sheet. A large amount of plastic straining was visually observed during the loading between the last photograph taken at 24,400 lbs. load and failure at 25,700 lbs. load.

Along every longitudinal line on the test specimen, the total longitudinal displacements between loading grips must have been the same at a given tensile load. These displacements appear as areas under the curves in the plots of longitudinal strains along longitudinal lines. Thus, longitudinal lines with relatively large strains near the transverse axis must also contain relatively small strains to equalize the areas under the strain curves. This tendency to equalize areas should be as shown in Figs 7B and 7G. The large maximum strain of Fig. 7B rapidly decreased to a value of less than 0.5% strain, while the 1.0% strain of Fig. 7G continued far in the longitudinal direction. Although the tendency for the strain to equalize displacements was indicated in most of the results, the action of the small strains in the elastic region can be established definitely only with increased experimental accuracy.

The flatness of the strain-longitudinal position curves at the lower strain values could cause great errors in the locations of the strain contours of Fig. 9. A more accurate strain map would have required more strain measurements from additional columns of dots, and higher measurement accuracy.

Improvement of Accuracy

Table 1 shows that the inaccuracies contributed by each group of error sources were of the same orders of magnitudes. Close observation of the dots on the specimen showed that "clearness of dot edge" and "sharpness of reproduction of dot edges on negative" are places where improvement may be expected. A commercial lithograph plate emulsion on the specimen would greatly improve sharpness.

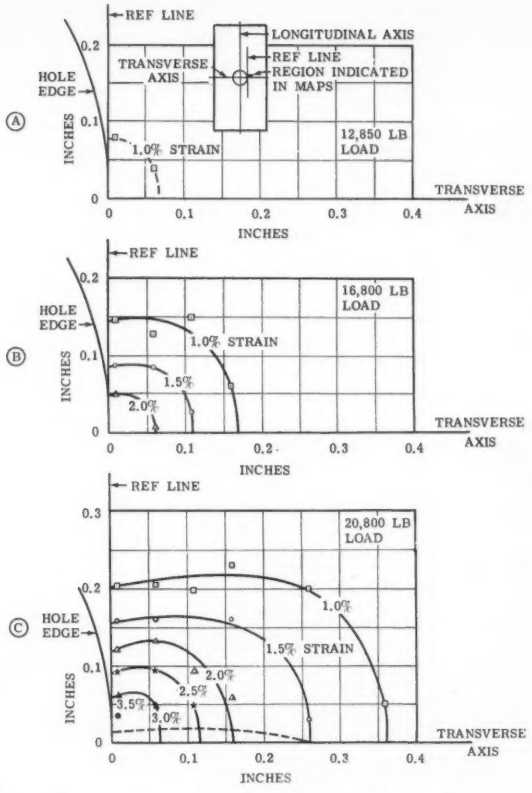


Fig. 9—Area of test specimen showing estimated locations of lines of constant strain

"Operator errors" could be reduced with binocular microscopes to reduce operator eye fatigue.

"Nonuniformity of light transmittance to negative causing changes in dot size," caused by light reflecting from the surface of the specimen, could be reduced by making the dots on the specimen negative rather than positive. This would leave the majority of the specimen blanked off with emulsion.

A microscope reticle with closely spaced lines would reduce the errors caused by dot-size changes. With this reticle, dots could be centered on the central hairline by aligning both sides of the dots with the appropriate hairlines. Only nonuniformities of dot-size changes would appear as strain errors.

A measuring device could be constructed which would view two dot-pattern negatives at once. In addition to allowing direct measurement of dot displacement, the device would permit compensation for dot-edge irregularities.

Conclusions

1. For loads less than 20,800 lbs.:

(*a*) The maximum longitudinal plastic strains along any longitudinal line were very close to the transverse axis.

(*b*) Longitudinal strains greater than 1.5% were within a very small region close to the hole edge at the transverse axis.

(*c*) Region A (Fig. 1) remained elastic.

2. Between a load of 20,800 lbs. and failure at a load of 25,700 lbs., strains greater than 1.5% spread rather extensively.

3. Strains apparently adjusted in the elastic region of the test specimen to cause the total longitudinal displacements through the hole edge to equal the total displacements far from the hole.

4. Discontinuities caused by slip (if they existed) were less than the order of magnitudes of the errors of the measuring system.

5. With the standard nameplate manufacturing process and the equipment readily available in a mechanics of materials testing laboratory, an estimated uncertainty interval of 0.3% strain with 95% confidence was achieved over a 0.050-in. gage length.

6. Improvements in accuracy could be expected without major changes in the experimental equipment or procedures.

Acknowledgments

The author is indebted to E. A. Davis of the Westinghouse Research Laboratories and W. R. Rohrer of the University of Pittsburgh for their able guidance. The work of R. J. Gregory and J. M. Bateman in adapting the nameplate manufacturing process to the production of the dot patterns is greatly appreciated.

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Photoelastic Study of Centrifugal Stresses in a Single Wheel and Hub

Paper by R. Guernsey was published in January 1961, issue of EXPERIMENTAL MECHANICS, pages 1–7

Discussion

by L. D. McConnell and H. Fessler

The author states that the object of the work is to "ascertain how accurately the stresses are determined by the modified plane stress theory." We should like to point out that the conclusions may be misleading. We believe that, in some circumstances, it may be dangerous to rely on the accuracy of modified plane stress theories.

Numerous tests which we have carried out independently confirm Dr. Guernsey's results that at the mid-plane the hoop stress at the bore is greater than the mean stress and that an axial compressive stress occurs in the same position. Thus the greatest shear stress is appreciably greater than would be predicted. In fact, the results from the paper show a maximum shear stress some 40% in excess of the calculated value. In other cases this may well be doubled. Fessler *et al.* show this in a recent paper* These factors can have a profound effect on both the initial yielding and the fatigue strength of a wheel.†

L. D. McConnell and H. Fessler are associated with Rolls Royce, Derby, U. K., and University of Nottingham, U. K., respectively.

** Fessler, Hay and Roberts, The Engineer, Jan. 27, 1961.*

Authors' Closure

The author agrees with Messrs. McConnell and Fessler that the modified plane stress theory used in the analysis fails to give a very accurate picture of the true stress distribution, especially in the mid-plane of the wheel. This was the sense of the entire paper and the corresponding conclusion was clearly stated. The geometry used in the test was deliberately designed to bring out the weaknesses of the plane stress calculations, and it was expected that such calculations would be accurate only in an average sense. This proved to be the case.

The existence of a compressive axial stress at the bore does indeed increase the maximum shearing stress at that point well above the nominally calculated value, and might have an effect on initial yielding. This fact, although implicit in the results, was not commented on specifically in the paper. The author wishes to thank the discussers for emphasizing the point.

† Discussion on paper by Heyman., J. Proc. Inst. Mech. Eng., 172, p 531 (1958).

sesa news

National Officers Nominated for 1961-62 Term

Names of candidates for the SESA elective posts of president, vice-president, and the executive committee have been announced by the National Nominating Committee.

Heading the list of proposed officers is Irwin Vigness of the Naval Research Laboratory who was nominated for the presidency. Dr. Vigness is currently serving as vice-president. Slated for the two-year term of vice-president is Charles C. Perry, associate professor of engineering mechanics at Wayne State University. Nominated to fill the three elective posts on the executive committee are: Raymond A. Gallant, director of the Engineering Laboratory, Chevrolet Motor Co.; Dimitri Kececioglu, of Allis-Chalmers Manufacturing Co.; and James L. Walmsley of Bethlehem Steel Co.

Continuing in office for the coming term is Walter W. Soroka, head of the Acoustics Laboratory, University of California, who, as vice-president, serves for two years. Daniel C. Drucker of Brown University, whose term as president ends this year, will automatically become a two-year member of the Executive Committee while Benjamin J. Lazan, immediate past president, will continue to serve his second year on the Committee. The office of treasurer, currently held by William M. Murray, is appointive.

The Nominating Committee consisted of Milton M. Leven, chairman; Edward Wenk, Jr., and Dr. Lazan, all past presidents.

According to the SESA By-Laws, "a member may also be nominated by written petition signed by at least twenty-five members and submitted to the Secretary at least sixty days prior to the opening of the Annual Meeting. Each nomination, if accepted, shall be placed on the election ballot." A ballot will be mailed to each member not less than thirty days prior to the Annual Meeting. The latter, designated as the First International Congress on Experimental Mechanics, will be held in New York City on November 1–3.

Biographical summaries on the background of the candidates, recommended to the membership by the Committee, follow.

PRESIDENT

Irwin Vigness

A very brief outline of the activities of Irwin Vigness includes several salient features. He undertook undergraduate and graduate studies at the University of Minnesota leading to a Master's degree from the Department of Electrical Engineering in 1931, and a Ph.D. from the Department of Physics in 1934. He remained at Minnesota as an instructor until 1939, when he entered the employ of the U. S. Naval Research Laboratory, Washington, D. C. At present he is head of the Shock and Vibration Branch.

Dr. Vigness has made a number of contributions published in the *Proceedings* of SESA. These have dealt mainly with shock, dynamic loading, vibrations and strain-gage characteristics. He is serving his second year as a vice-president of SESA.

Irwin Vigness

VICE-PRESIDENT

Charles C. Perry

Hailing from Grand Blanc, Mich., C. C. Perry attended General Motors Institute where he received his diploma in automotive engineering while working alternate months with the Pontiac Motor Division. He served two years with the U. S. Navy as an electronics technician. Graduating from Wayne State University in 1947 with a B.S. degree in mechanical engineering, he joined the Central Engineering section of General Motors, first as design engineer and later as project engineer. During this time Prof. Perry earned his master's degree in engineering mechanics from Wayne State. From 1951 to 1953 he was with Vickers, Inc., as project engineer and then as engineer in charge of research.

Returning to Wayne State University in 1953, Prof. Perry became associate professor of engineering mechanics and established a private practice as a research consultant—both of which are his current vocations. During the summers of 1957 and 1960 he attended the AEC Institutes of Nuclear Science and Engineering at Argonne National Laboratory.

In the field of publications, he is co-author of the book, *The Strain Gage Primer* and editor of the SESA *Manual on Experimental Stress Analysis Techniques.* He has published numerous papers in engineering society journals, including SESA, and many articles in trade journals. He is presently working as co-author on a chapter on strain gages for the forthcoming *Handbook of Shock and Vibration Control.* He was also editor of the SESA *News Letter.*

A registered professional engineer in the states of Illinois, Indiana, Michigan and Ohio, he holds membership in the ASME, ASEE, ARS, Sigma Xi, Tau Beta Pi and in the SESA where he has held numerous local and national offices.

EXECUTIVE COMMITTEE

Raymond A. Gallant

Raymond A. Gallant is director of the Engineering Laboratory at the Chevrolet Engineering Center, Warren, Mich. He is responsible for all laboratory testing of passenger car and truck components, and for the building of all engines, axles and transmissions at the Engineering Center.

A native of Plattsburg, N. Y., Mr. Gallant graduated from Rensselaer Polytechnic Institute in 1950, and began his career with General Motors as a college graduate in training at the GM Research Staff. Taking an educational leave of absence in the fall of 1954, he spent one year doing graduate work, teaching and research on ball bearings at Rensselaer. He rejoined General Motors in 1955 when the new Chevrolet Engineering Center was opened, and organized the Laboratory Instrumentation Department, serving as head of that department until February 1961. At that time he was appointed assistant director of the Engineering Center Laboratory. He was promoted to his present position on Apr. 16, 1961.

Raymond Gallant received the B.M.E. degree from Rensselaer Polytechnic Institute in 1950, and the M.S. degree in engineering mechanics from the University of Michigan in June 1954. He is a member of the SAE, ASEE and a past chairman of the Detroit Section of SESA. He is currently Membership Chairman of SESA.

Mr. Gallant is also a member of Tau Beta Pi, Sigma Xi, Pi Tau Sigma and Alpha Psi Omega honorary societies. During World War II he served as a naval aviator, and now holds the rank of Commander in the Naval Reserve.

PROCEEDINGS MAILED

Proceedings of SESA, Vol. XVIII, No. 1, were mailed at the end of June to members who were in good standing as of June 30, 1960. Volume XVIII, No. 2, will be mailed at the end of this year to members who were in good standing as of Dec. 31, 1960. If any one in the first category did not receive his copy, please write SESA Headquarters, 21 Bridge Square, Westport, Conn.

Dimitri Kececioglu

Dimitri Kececioglu received his bachelor of science degree in mechanical engineering from Robert College, Istanbul, Turkey, in 1942, his master of science degree in industrial engineering from Purdue University, Lafayette, Ind., in 1948, and his doctor of philosophy degree in engineering mechanics also from Purdue in 1953. While at Purdue he taught theory of plasticity, and statics and dynamics, among other subjects.

In 1952, he joined the Allis-Chalmers Manufacturing Co. in Milwaukee, Wis., as an engineering scientist-in-charge of the Mechanical Laboratories in the Research Division. In 1957 he was made the assistant to the director of mechanical engineering, Industries Group, and, in August 1960, a consulting engineer, Staff Engineering, Industries Group of Allis-Chalmers. Since November 1960, he has been director of reliability.

He has presented five papers and written five technical articles. He has prepared a "Bibliography on Plasticity—Theory and Applications," published by ASME and he is a regular contributor of critical reviews to "Applied Mechanics Reviews." He holds five patents.

He is president-elect of the Milwaukee Technical Council, a director of Engineers' Society of Milwaukee, past chairman of the Society for Experimental Stress Analysis—Milwaukee Section—and is presently chairman of the ASME-Milwaukee Section. He also belongs to AAAS, ASEE and is a Tau Beta Pi and Sigma Xi. He is a member of the Papers and Proceedings Committee of SESA.

James L. Walmsley

James L. Walmsley was graduated from Alabama Polytechnic Institute in 1926 with a B.S. degree in civil engineering. He was employed by Bethlehem Steel Co. in 1926 as a bridge engineer. In 1943, he was transferred to the Research Department where he was concerned with structural and engineering mechanics problems. He now is supervisor of stress analysis and testing in engineering mechanics in the Research Department.

He is a member of the American Society of Civil Engineers, the Society for Experimental Stress Analysis and the American Welding Society. He was formerly a member of the Local Sections Committee of SESA and has served on the Weld Stress Committee of the Welding Research Council and on the joint committee on mill rolls research of the Association of Iron and Steel Engineers-Rollmakers' Institute.

Mr. Walmsley is a registered professional engineer in Pennsylvania and lives in Bethlehem.

A Note to AUTHORS

- All papers and technical notes should conform to style requirements set forth in the manual "An SESA Paper" and the "Authors' Check List." Obtain your copies from SESA Headquarters, 21 Bridge Square, Westport, Conn.
- All papers and technical notes must be reviewed and accepted by the Papers and Proceedings Committee in order to qualify for publication.
- Authors must provide a statement that their articles have not been previously published nor submitted for publication elsewhere.
- One original and two copies of the manuscript, complete and carefully typed with double spacing and wide margins, are to be sent to: Roscoe Guernsey, Chairman, Papers and Proceedings Committee, Space Technology Laboratories, Inc., P. O. Box 95005, Los Angeles 45, Calif.
- Authors should state their affiliation, title or position and mailing address.
- All papers, but not necessarily technical notes, must be presented at a Society Meeting either orally or by title.
- If at all possible, length of text should not exceed 4000 words; a technical note should not contain more than 750 words.
- Illustrations must be clearly drawn in ink and should not be larger than $8\frac{1}{2}$ x 11 in. "Guided" lettering should be used at all times and should be large enough to be legible when reduced. Photographs must be sharp, glossy prints. A separate list of captions for all figures must be typed with double spacing. The figure number and author's name should appear on the back of each illustration.
- Reprints may be ordered from SESA Headquarters.

Papers by Foreign Speakers to Highlight First International Congress

Authors from a half-dozen countries in three continents will join their American colleagues in presenting technical discussions when the First International Congress on Experimental Mechanics convenes November 1–3 at the Hotel New Yorker in New York City. Bringing with them the results of investigations carried on overseas, the authors from abroad will also present national survey reports to make the coming Congress a world-wide forum on present-day activities in the field.

The three-day Congress will be a complete SESA affair including the important manufacturers' equipment exposition, educational exhibits, students and advanced seminars, social activities, ladies program, the William M. Murray Lecture, Society business and committee meetings and, of course, the technical sessions. As present plans are shaping up, the latter will probably be divided into five daytime sessions and one night session. The W. M. Murray luncheon and lecture will be a separate event.

Among the invited foreign speakers, two will present survey papers and five will present the results of investigations. An additional paper will be by title only. The survey papers will be by D. G. Sopwith of the National Physical Laboratory, East Kilbride, Glasgow, Scotland, who will present "Experimental Mechanics at the National Physical Laboratory," and R. Boiten of the Institute TNO, Delft, The Netherlands, who will discuss "Experimental Mechanics at TNO." The following list gives the names of foreign authors who will present studies, their affiliations, country and the title of their papers:

P. Dantu, Laboratoire Central des Ponts et Chausses, Paris, France—"Moiré Effects at High Temperature."

V. M. Hickson, Ministry of Aviation, Royal Aircraft Establishment, Farnborough, England—"Special Techniques in Experimental Stress Analysis."

E. Hoek, South African Council for Scientific and Industrial Research, Pretoria, South Africa—"Experimental Study of Rock Stress Problems in Deep Level Mining."

A. Mellgren, Industrial Reactor Division, Aktiebolaget Atomenergi, Stockholm, Sweden—"Measuring Creep at Small Strains."

G. Oberti, Istituto Sperimentale Modelli E Strutture, Bergamo, Italy—"Experimental Studies on Structures Beyond the Elastic Limit."

L. G. Randell, Aerodynamics Division, Aeronautical Research Laboratories, Melbourne, Australia—"An Experimental Self-temperature-compensated Strain-gage System" (by title).

At least fifteen additional papers from outstanding sources in this country and abroad will present a great variety of topics. Some idea of the scope may be gained from a descriptive listing of a number of the papers now under consideration. Among these are: "determination of slope contours in flexed elastic plates by the Salet-Ikeda technique"—"experimental analysis on the strength of cylindrical shell under external pressure"—"experimental mechanics in the development of a miniature pressure transducer"—"the Micro-Sensor—a new tool for experimental stress analysis"—"experimental investigation of eccentric longitudinal impact of bars"—"experimental investigation of wave propagation in isotropic elastic plates"—"large inelastic deformation of a circular membrane under static and dynamic loads"—"model studies of a radome shell under simulated wind loading"—"analysis of two-dimensional stress problems by the Moiré method"—"transient behavior of rocket sleds"—"new method for determining the elastic compliances of stiffened plates."

A complete schedule of all events for the three days will appear as a tentative program in the September issue of EXPERIMENTAL MECHANICS. Additional information of interest will appear regarding some of the outstanding special events now being arranged.

Application for Corporate Membership

SESA

Society for Experimental Stress Analysis

Name of Company _____

Name of Representative _____

Position or Title _____

Department _____

Number and Street _____

City _____ Zone _____ State _____

We wish to apply for Corporate Membership in the Society for Experimental Stress Analysis for the period indicated below:

_____ July 1, 1961 through June 30, 1962 Dues: $100.00

We understand that we are to receive the full benefits prescribed by the Society for the designated period.

Date Authorized Signature _____

_____ Title _____

(Please give billing address, if different from above. Mail completed application to: Executive Secretary, Society for Experimental Stress Analysis, 21 Bridge Square, Westport, Conn.)

Tentative Technical Program

for the

International Symposium on Photoelasticity

October 29–31, 1961

Illinois Institute of Technology

Chicago, Illinois

Sponsored by Illinois Institute of Technology, Office of Ordnance Research of the U. S. Army, National Science Foundation, Society for Experimental Stress Analysis, Office of Naval Research, Office of Scientific Research, U. S. Air Force, Applied Mechanics Division of the American Society of Mechanical Engineers, and the Chicago Sections of the SESA, ASME, and ASCE.

Sunday, October 29

SESSION I—2:00 P.M.

Birefringent Coatings

Recent Activity in Photoelasticity in Japan,
Ziro Tuzi, Riken Keiki Fine Instrument Co., Ltd., Tokyo, Japan

An Experimental Study of the Effects of Thickness of Birefringent Coatings,
J. Duffy and C. Mylonas, Brown University, Providence, R. I.

An Investigation of the Elasto-plastic Strain Distribution Around Cracks in Various Sheet Materials,
J. R. Dixon and W. Visser, National Engineering Laboratory, East Kilbride, Glasgow

Photoelastic Coating Analysis in Thermal Fields,
F. Zandman and S. S. Redner, The Budd Company, Phoenixville, Pa.
D. Post, Rensselaer Polytechnic Institute, Troy, N. Y.

Elasto-plastic Stress Analysis and the Determination of Flow Lines by Means of Photoelastic Coating Method,
K. Kawata, Aeronautical Research Institute, University of Tokyo, Japan

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Monday, October 30

SESSION II—9:00 A.M.

Photothermoelasticity and Photoplasticity

Progress in Photothermoelasticity,
G. Gerard, New York University, New York, N. Y.

A Study of the Accuracy and the Application Limits of Plane Photoplastic Problems,
E. Monch and R. Loreck, Technische Hochschule, Munchen, Germany

An Experimental Study of the Laws of Temporary Double Refraction in the Plastic State in Cellulose Nitrate—Foundations for Three-dimensional Photoplasticity,
M. M. Frocht and Y. F. Cheng, Illinois Institute of Technology, Chicago, Ill.

Monday, October 30

SESSION III—2:00 P.M.

Two- and Three-dimensional Photoelasticity

Development of the Lateral Extensometer Method in Two-dimensional Photoelasticity,
R. Hiltscher, Kungl. Vattenfallsatyrelsen, Stockholm, Sweden

A Photoelectric-interferometric Method to Determine Separately the Principal Stresses in Two-dimensional States and Possible Application,
H. Favre, and W. Schumann, Ecole Polytechnique Federale, Zurich, Switzerland

Photoelastic Determination of Shrink-fit Stresses in Three-dimensional Models,
R. Guernsey, Jr., General Electric Co., Schenectady, N. Y.

Epoxy Resins for Photoelastic Use,
M. M. Leven, Westinghouse Research Laboratory, East Pittsburgh, Pa.

Tuesday, October 31

SESSION IV—9:00 A.M.

Dynamic Photoelasticity and Three-dimensional Photoelasticity

Stress Distribution on the Boundary of a Square Hole in a Large Plate During Passage of a Stress Pulse of Long Duration,
A. J. Durelli, Illinois Institute of Technology, W. F. Riley and J. Carey, Armour Research Foundation, Chicago, Ill.

A New Two-layer Technique for the Photoelastic Analysis of Loaded Plates,
Ch. H. Lerchenthal and A. A. Betser, Israel Institute of Technology, Haifa, Israel

New Photoelastic Methods for Torsion Problems,
M. Nisida, The Institute of Physical Chemical Research, Tokyo, Japan

A Three-dimensional Photoelastic Investigation of a Propeller Blade Retention,
M. M. Frocht, Illinois Institute of Technology, Chicago, Ill., and B. Wang, International Business Machines Corp., Poughkeepsie, N. Y.

section reports

Great Salt Lake

Strain-gage Properties and Applications

The fifth meeting of the *Great Salt Lake* Section was held on May 24 at the New Engineering Building, University of Utah. Thirty-six members and guests attended the business meeting and the talk given later by Jim Starr. A distinguished guest of the evening was William Bean.

Regarding the business meeting, Chairman C. S. Barton opened the meet- and announced our acceptance into the national SESA. Summer meetings were discussed, and it was decided to see what the program committee could arrange for July. George West urged those present to recommend corporate memberships to their companies. Al Skinner announced that we had been accepted into the Utah Engineering Council, and he noted the aims and objectives of the council. The drive for national and local dues payments was renewed. The Chairman also noted that he had urged consideration of Salt Lake City as a site for a future national convention.

For the Technical talk, Denny Henneman introduced the speaker of the evening, Jim Starr, presently director of engineering and research at Allegany Instrument Co. who spoke on "Strain Gages." Mr. Starr reviewed the history of gage development from the carbon graphite gage through the semiconductor gage. He touched on adhesive bonding and wire or foil support and discussed changes in strain sensitivity vs. gage factor. Since wire-wound gages are handmade, they are not as repeatable as foil-type gages.

Resistance-type strain gages are most widely used in transducer work because resistance impedance is not frequency sensitive, such as induction or capacitance. Semiconductor gages, with their high output will replace inductance and capacitance-type transducers. However, the cost of changing present equipment and other circuit difficulties, as well as their more rigid handling requirements, will limit the use of semiconductor gages in experimental stress work, at least in the immediate future. The high output will be useful in slip ring and low strain applications.

Transducers usually involve special mechanical springs with strain gages attached. Accuracy requires special care and precision in mounting gages. A full bridge should be on the transducer to minimize temperature and lead-wire effects. Temperature-compensated gages improve stability if properly matched.

(Reported by S. McClure for Great Salt Lake Section)

Enthusiasm Key to Growth at Great Salt Lake

The recent formation of the *Great Salt Lake* Section came about as the result of enthusiasm and hard work, according to Cliff S. Barton, newly elected chairman. In describing recent activities that led to their recognition as a unit of the SESA, Dr. Barton said:

"I should like to hasten to assure the national office that we have a great potential for SESA work in the State of Utah and our national membership should grow rapidly. I believe I am safe in making this statement since the activities of this past year will bear it out. Through the cooperative efforts of three major universities, Brigham Young University, University of Utah, Utah State University and several major industries such as Thiokol Chemical Corp., Hercules Powder, etc., we started from scratch last October and carried forth four very successful section meetings with the fifth one being held on May 24, 1961. Our fourth meeting was attended by Mr. Clarence Smith, the National Chairman for Local Sections, who was our speaker of the evening."

As a further indication of activities in Utah, Dr. Barton added, "We have organized a Membership Committee and a Program Committee with John McCrosson as chairman of the Program Committee. Our membership has indicated a desire to continue meeting throughout the year. We are planning on doing this instead of taking a summer vacation. The enthusiasm is so high that we are submitting a petition to hold a future annual meeting in Salt Lake City."

Newly-elected officers worked hard to establish the Great Salt Lake section. They are, left to right, first row, C. S. Barton, chairman; George West, vice-chairman; Vance Christiansen, advisory board; E. W. Matrejek, advisory board; (back row) Don Mayhew, treasurer; Wayne S. Brown, advisory board; John McCrosson, program chairman; Alfred Skinner, advisory board. Not shown is S. R. McClure, secretary.

Northern California

Be Active—
Support Your Society

Southwest

Automobile- barrier Impacts; Stress Analysis of Rockets

For the second time in three years the Aerojet Corp. of Sacramento was the gracious host for a very full *Northern California* Section meeting. The four highlights of the May 1961 meeting were: (1) a presentation by the California State Highway Department on dividing-strip barrier impacts and measurement of highway deflections; (2) a review of experimental stress analysis and testing techniques for solid-rocket pressure vessels and liquid-rocket engines; (3) a tour of the solid-rocket plant experimental test laboratory; and (4) a tour of the Titan missile engine plant.

W. Weber introduced Mr. Fields who showed high-speed motion pictures of automobiles impacting various types of dividing-strip barriers at speeds of 60 mph and an angle of incidence of 60 deg from the normal. These results indicated that too stiff a barrier caused a deceleration too great for passenger safety, barriers too low allowed the auto to ramp over the barrier and an elastic barrier caused the auto to rebound back into the traffic lanes. The most successful barrier was a 5-ft high chain link fence with conventional steel posts using for longitudinal members three $^1/_2$-in. steel cables—one at the top of the fence and two half-way up—one on each side of the posts. Cars or buses striking this barrier produced a large deflection of the fence, demolishing several posts and the included chain link section. However, the longitudinal cables held and trapped the car so that it did not rebound or cross the dividing strip. Tolerable occupant decelerations were recorded.

Les Hanibal showed pictures of a "leap-frogging" device carried by a heavy semitruck-trailer for obtaining almost continuous recordings of a highway under actual low-speed loading conditions. Thousands of feet of highway could be checked in one day by driving the truck at a steady speed of 1 mph to produce the loading while the deflection measuring rig was automatically lifted up, moved forward several feet and set down again.

R. K. Steele outlined some of the problems in testing solid propellent tanks under pressure, with and without a fiberglass covering. One breakthrough described was the keyed solid propellent tank joints which would support 22,000 lb per lineal inch and allow for quick assembly of very long tanks at launching sites. R. C. Sampson continued with a summary of photoelastic tests on various shapes of solid rocket propellent grains and stress distribution in the same keyed tank joints.

R. Sanders then conducted the group through the Experimental Test Laboratory of the Solid Rocket Plant. An actual solid propellent tank was failed hydraulically for the group and an interesting acoustic recording of the noise produced by the slipping fiberglass cover was displayed. The hardware involved in strain gaging and making deflection and photoelastic measurements was displayed by John Zickel's group.

After a much appreciated dinner in the plant cafeteria, several Aerojet personnel conducted the group through the Titan missile engine production plant. Many unconventional problems and techniques were witnessed. Regenerative cooling tubes were furnace welded. Austempering was used to minimize distortion and to speed up production. Tape-programmed automatic drilling machines were used to drill the myriad multiangled holes. Furnaces on tracks can heat entire missile fuel tanks and move them over equally large quenching tanks.

The evening was culminated by a bus tour of the static rocket firing facilities. Once again the Northern California Section almost witnessed an actual static firing.

Much credit is due Al Hardrath and John Zickel and others for arranging this very worth-while meeting.

(Reported by D. M. Cunningham for the Northern California Section)

Simon Rodia's Towers

The Annual Meeting of the *Southwest* Chapter was held on Apr. 20, 1961. The meeting was called to order by the chairman, G. D. Lindeneau. The election of officers for the Southwest Chapter was held. The following men were elected to office by a majority vote of the membership: chairman, Sanford Friezner; vice-chairman, A. R. Vollmecke and director for two years, James W. Bice.

Mr. Lindeneau, as outgoing chairman, expressed his thanks to all those that had worked in the chapter under his administration. It is noted that the chapter membership had increased 15% and now is 130 members.

Following the business meeting, two papers were presented. The first was by W. E. Witzell of the Experimental Stress Analysis Laboratory, Convair-Astronautics. Mr. Witzell spoke on "Some Experiments with Photoelastic Coatings." He discussed some of the problems involved with a spray coating technique which has been successfully used at Convair-Astronautics. His talk was accompanied by color slides showing sprayed photostress on spot-welded joints.

N. J. Goldstone spoke on "The Static Test of Simon Rodia's Towers." Mr. Goldstone stated that his test of the towers was not part of his normal duties as a test engineer. Mr. Goldstone discussed the history of the Rodia Towers. Simon Rodia was, by trade, a tile-setter and worked for 33 years to build a series of open-frame towers on a triangular piece of land by the railroad track in Watts, a community in southwest Los Angeles. The towers,

SOUTHWEST PLANNERS MEET

Having completed plans for the Annual Equipment Exhibit of the Southwest Section, the committee in charge smiles a cordial welcome. Left to right, J. Ferentz; Exhibit Chairman E. W. Larson; Section Membership Chairman B. J. Wolfe; Section Chairman S. Friezner; and L. Berman

some up to 100 ft high, made of steel and concrete, and each inlaid with tile, broken glass and sea shells, were built alone by Rodia. The Watt's Towers are recognized as a unique expression of art by authorities throughout the world. In 1959, the towers were declared dangerous by the City Building Department and ordered to be demolished. The threat of demolition aroused local and national protest from those who knew of the towers. Mr. Goldstone became interested in the towers in reading the structural analysis made by the city and became convinced that this analysis was in error. He was later approached by a committee of citizens who asked for technical help in protecting the towers. The result was a test simulating wind forces applied to the tallest tower. The test was performed before a television camera, art authorities, architects and just plain people. The tower deflection and strains were monitored while eleven assistants including a UCLA professor and architects worked studying the bending of the towers through binoculars. Simon Rodia was shown to be as much or a structural engineer as an artist in that the towers successfully withstood the test. Mr. Goldstone's talk was accompanied by a 20-min sound movie of the test.

The next regular meeting of the Southwest Chapter is scheduled for September.

(Reported by E. K. Winslow for the Southwest Section)

Washington Area

Vibration Symposium Scores Big Success

The National Bureau of Standards on May 4 played host to 161 persons attending the all-day Symposium-Workshop on Vibration, Its Measurement and Control. Planned primarily to effect a cross-pollenation of ideas from different branches of vibration technology, the symposium was a joint undertaking by the *Washington Area* Section of SESA and the local chapter of the Instrument Society of America.

In his welcoming remarks, W. A. Wildhack, special assistant to the Director of NBS, affirmed the importance of such scientific exchanges, and reasserted the Bureau's desire to encourage such events, which are so closely related to that agency's mission.

The success of the meeting is believed to have stemmed largely from the efforts of the planning committee under Roscoe L. Bloss of NBS, whose decision not to publish the proceedings of the symposium elicited presentations of very recent developments from the speakers. Other members of Mr. Bloss' committee were Victor Dawson of the Naval Ordnance Laboratory, Ralph G. Barclay of the Diamond Ordnance Fuze Laboratories and John P. Kearns of Johns Hopkins University's Applied Physics Laboratory.

Mr. Kearns, incidentally, served double duty. As one of the symposium's seven invited speakers, he also discussed and demonstrated "Flutter Simulation" in a sequel to the paper he presented at SESA's 1959 Spring Meeting in Washington.

One of the most interesting presentations of the day was a discussion of "Man-Made Vibrations in Buildings and in the Ground" by Douglas R. Tate of NBS. Mr. Tate's paradoxical conclusion was that locations for new laboratory buildings can be found which are relatively free from man-made vibrations, but that this advantage disappears when the laboratory has been constructed and occupied.

The final paper of the symposium, by Harold M. Forkois of NRL, was perhaps the most directly useful of all. His talk, "Designing for Shock and Vibration," was a discussion of the basic considerations for structural design to resist shock and vibration.

(Reported by L. Mordfin for the Washington Area Section; photographs by N. Halsey)

Horace Trent presents the opening paper "Some Pitfalls to be Avoided in the Recording and Analysis of Vibrations"

John C. New, chairman of the morning session, reviews the need for straightforward procedures for handling certain types of vibration problems

Ben Reznek emphasizes the concept of system calibration as differentiated from component calibration, in his paper "Calibration of Accelerometer Systems"

Father Bernard McConnel summarizes the state of knowledge of "Seismic Vibrations"

Erich Buchmann compares "Calculated and Measured Vibration on NewlyDesigned Ship," as I. Vigness, chairman of afternoon session, stands by

from the campus

Gordon McKay Laboratory of Applied Science, Harvard University

EQUIPMENT EXHIBIT TO BE HELD WITH UCLA COURSES

The Fourth Annual Strain Gage Equipment Exhibit sponsored by the *SESA Southwest Section*, is to be held on Aug. 31, 1961, in Los Angeles, Calif.

The exhibit is being held to coincide with the three short courses on Strain Gages and Photoelasticity which are held annually at the University of California at Los Angeles. All of the faculty, engineers and students participating in these courses will be invited to the exhibit and luncheon free of charge.

The exhibit, to be held in the Satellite Room of the Miramar Hotel in Santa Monica, will start with a luncheon at 12:30 P.M. followed by several very short speeches by the exhibitors. Immediately thereafter, the exhibit areas and the bar will be open until the close of the exhibit at approximately 5:30 P.M.

Attendance is by invitation only and is completely free of charge. This includes the luncheon, cocktails and admission to the exhibit areas. Invitations will be sent to all paid members of the Southwest Section and participants in the strain gage courses. Guests will include members of the Western Regional Strain Gage Committee and representatives from west coast companies interested in experimental mechanics. Companies interested in sending people to the exhibit should contact the committee for further information. It is expected that approximately 300 people working in the field of experimental mechanics will be present. Members from other chapters will also be welcomed, free of charge, if they will notify the committee of their plans to attend.

Most of the manufacturers of strain gages, strain gage instrumentation, load cells, accelerometers, pressure transducers, photoelastic equipment, recording devices and other equipment useful in experimental stress analysis work will be represented. A special exhibit of textbooks and other literature on stress analysis is being included for the first time.

Exhibit space in the form of a 3 x 8 ft table and suitable electric power will be provided for each exhibitor. This space can be rented for $75 and companies interested in this exhibit should write or phone: Barry J. Wolfe, P.O. Box 281, Northridge, Calif., Phone—Los Angeles, TRiangle 7-5135.

Reservations for exhibit space will be made on a first-come, first-served basis.

WAYNE STATE TO SPONSOR STRESS ANALYSIS INSTITUTE

A two-week special institute in experimental stress analysis will be given by the Engineering Mechanics Department of Wayne State University from Sept. 11 through Sept. 22, 1961. Joseph Der Hovanesian, the institute coordinator, announced that the institute will be offered in two programs: Program 1—The Theory and Application of Photoelasticity—will be offered during the week of Sept. 11, 1961. Program 2—Strain-gage Theory and Application—will be given during the week of Sept. 18, 1961. Students may elect to enroll for both programs for two weeks at a tuition of $325, or for either one of the programs for one week at a tuition of $175.

The staff of 15 or more lecturers will be drawn from Wayne State University and from various parts of the country.

The institute will be conducted over a full eight-hour day with some optional evening sessions. The time between lecture and laboratory work will be evenly divided. Essentially, all classical experimental stress analysis topics will be presented with special emphasis on latest developments. A certificate will be awarded to participants upon satisfactory completion of one or both programs of the institute.

SUMMER COURSE IN METROLOGY

George Washington University's Center for Measurement Science will offer, from August 14 to 25, a special course on the science of precise measurement.

The Center, established last fall with the support of the National Bureau of Standards and the Martin Co. of Baltimore, Md., was formed to meet a vital gap in America's scientific educational program. The first classes were held in February at the University campus.

A complete undergraduate and graduate schedule of study is offered now at the Center, which is the first of its kind in the United States. Advanced training, in-plant study courses and special programs are planned or under way.

The summer course will program a full schedule of classes from 8:30 A.M. to 4:15 P.M. daily, Subjects to be covered include "Design of Experiments" and "Time and Frequency Measurements."

Instructors for the classes will be drawn from the George Washington University faculty, the National Bureau of Standards, Johns Hopkins Applied Physics Laboratory and the Weinschel Engineering Co. Laboratory sessions will be under the auspices of the Bureau's educational committee.

NEW CURRICULUM OFFERED

The University of Wisconsin faculty has approved a new curriculum leading to the degree of bachelor of science, nuclear engineering.

Undergraduate students as well as graduate students will be able to study nuclear engineering in the UW College of Engineering beginning next fall. Under the new curriculum, from 40 to 44% of the student's studies will be in the basic sciences, also 40 to 44% in engineering, and from 16 to 20% in nontechnical courses.

The UW College of Engineering's nuclear engineering program was started three years ago for graduate students only, granting master of science degrees only during its first two years. The program was enlarged a year ago to grant also Ph.D. degrees.

Three Short Intensive Courses on

STRAIN GAGES & PHOTOELASTICITY

at the DEPARTMENT OF ENGINEERING
UNIVERSITY OF CALIFORNIA AT LOS ANGELES

FOR WHOM INTENDED

These three courses have been designed to present an integrated program on theory and practice, including a review of fundamentals and discussions of some of the latest techniques, especially those suited to high and low temperature strain gage applications. Attention will also be given to instrumentation, which, in many cases, can be used equally well with other kinds of transducers.

The courses are intended primarily for those with a bachelor's (or higher) degree in engineering, physics, or metallurgy. However, in certain cases, experienced technicians without degrees can be expected to gain full benefit.

The material content of the courses, although presented largely from the point of view of the mechanical engineer, has been selected to provide maximum benefit for those who are already skilled in stress analysis but who may have little experience in electrical instrumentation, and likewise for electronics experts who may not be quite so familiar with certain aspects of stress analysis.

STRAIN GAGE LECTURE COURSE

August 21 - 25, 1961 (Monday through Friday)

The first of the two strain gage courses will consist of a series of lectures supplemented with illustrative demonstrations. Some of these lectures have been prepared especially for beginners and consequently are of an elementary nature; however, other lectures of a much more advanced character will also be included in the series. The latter group will contain detailed information, especially relating to high temperature strain gages, which will be useful to those who already possess considerable experience. The content of the course has been selected and organized for maximum benefit to designers, metallurgists, laboratory workers, laboratory supervisors, theoretical stress analysts, and others, who wish to advance their knowledge relating to the uses of strain gages and the techniques which are available for producing optimum results.

Both the theoretical and practical aspects of strain gages themselves, methods of installation, and allied equipment, will be presented in the lectures. Details of strain gage applications for measuring force, torque, bending, shear, and accelerations, will be discussed as well as simple electric circuits for automatic computation of quantities which depend upon the simultaneous observation of strains at more than one location.

The following will be included in the topics which are to be presented:

Basic theory of Wire, Foil, and Semi-Conductor strain gages
Details of gage characteristics
Techniques for installing gages
Mechanical and Electrical aspects of the gage system
Fundamental Laws of Measurement
Electric Circuits for strain gages
Dynamic and Static Strain Measurements
Computing Bridge circuits for direct measurement of quantities depending upon several strains
Stresscoat as an Auxiliary Tool
Measuring systems and instruments
Low-temperature applications of strain gages
High-temperature strain gages and their installation
Commercial Equipment
Rosette analysis and interpretation of observations

STRAIN GAGE LABORATORY COURSE

August 28 - September 1, 1961 (Monday through Friday)

For those who wish to gain personal experience in the installation of gages and the handling of equipment, a second course containing a series of laboratory experiments will be offered. The laboratory program is supplementary to the lecture course. It will consist of a set of projects, each containing one or more experiments, which have been designed to illustrate the topics taken up in the lecture course during the previous week. The following exercises will be included to provide practice in the practical applications of strain gage techniques:

Strain Gage Installation for normal and high temperature applications
Static observations with single and multiple gage bridges
Dynamic strain Measurements
Characteristics and checking of electrical equipment used with strain gages
Strain Rosette Analysis
Characteristics and handling of certain commercial equipment
Measurement of Force, Torque, Bending, and Moments at inaccessible locations

Since the benefits which can be obtained by attendance at the laboratory program depend to a large extent upon an understanding of the underlying theory, registration in the laboratory course will be limited to those who have attended the previous lecture course, or its equivalent in prior years.

INDUSTRIAL PHOTOELASTICITY

(Lectures & Laboratory)

September 5 - 9, 1961 (Tuesday through Saturday)

The course in Industrial Photoelasticity will review the fundamental theory and laboratory techniques related to the classical applications of polarized light for solving stress problems. In addition to this, discussions of some of the most advanced techniques, including Three-Dimensional Photoelasticity and Photoelastic Surface Coatings (PhotoStress), will also be presented.

The course will emphasize the ways in which photoelasticity can be used to solve industrial problems. Attention will also be given to the advantages and limitations of the method and its relation to other experimental techniques with which it may often be combined most effectively.

The material will be presented through lectures, demonstrations, and laboratory exercises. Consideration will also be given to the properties and selection of photoelastic materials as well as the specification of equipment required to set up a laboratory. This program has been designed especially for the engineer or metallurgist who has little or no previous knowledge of the subject.

TUITION

The tuition per registrant for each course will be as follows:—

Strain Gage Lecture Course	**$175.00**
Strain Gage Laboratory Course	**$175.00**
Industrial Photoelasticity	**$200.00**

FOR FURTHER INFORMATION, call or write,

ENGINEERING EXTENSION, DEPARTMENT OF ENGINEERING,
Room 6266, ENGINEERING-PHYSICAL SCIENCES BUILDING,
UNIVERSITY OF CALIFORNIA, LOS ANGELES, 24, CALIFORNIA
Telephone — GRanite 3-0971, Extension 369

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